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RESEARCH MEMORANDUM

CORRELATION OF CYLINDER-HEAD TEMPERATURES OF A
MULTICYLINDER, LIQUID-COOLED ENGINE
OF 1710-CUBIC-INCH DISPLACEMENT

By Bruce T. Lundin, John H. Povolny, and Louis J. Chelko

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SUMMARY

Data obtained from an extensive investigation of the cooling characteristics of four multicylinder, liquid-cooled engines have been analyzed and a correlation obtained between the cylinder-head temperatures and the primary engine and coolant variables. The method of correlation was previously developed by the NACA from an analysis of the cooling processes involved in a liquid-cooled-engine cylinder and is based on the theory of nonboiling, forced-convection heat transfer. The present correlation, which covers engine power outputs up to 1860 brake horsepower, coolant flow rates from 50 to 320 gallons per minute, and coolants composed of ethylene glycol - water mixtures varying in composition from 100-percent water to 97-percent ethylene glycol and 3-percent water, included ranges of engine speed, manifold pressure, carburetor-air temperature, fuel-air ratio, exhaust-gas pressure, ignition timing, and various coolant temperatures. The effect on engine cooling of scale formation on the coolant passages of the engine and of boiling of the coolant under various operating conditions is also discussed.

The results of this analysis indicated that the correlation method is applicable to multicylinder, liquid-cooled engines of the type investigated and permits the prediction of the cylinder-head temperature between the exhaust valves within approximately $\pm 12^{\circ}$ F for any operating condition within the range of the investigation.

INTRODUCTION

An investigation of the cooling characteristics of reciprocating aircraft engines is of importance in order to insure satisfactory engine performance at extreme conditions of operation.

A considerable amount of data on the cooling characteristics of various air-cooled engines have been published by various investigators but little data have been published on the cooling characteristics of liquid-cooled engines.

An extensive research program to determine the cooling characteristics of liquid-cooled engines was therefore instituted at the NACA Cleveland laboratory in 1943. The initial phase of this program consisted of an investigation conducted on a single-cylinder engine to provide data for a fundamental study of the heat-transfer processes involved. The final results of this investigation are reported in reference 1 in which an analysis, based on the theory of nonboiling forced-convection heat transfer, was made of the cooling processes in a liquid-cooled engine. This analysis resulted in a semiempirical method, similar to that presented in reference 2 for air-cooled engines, of correlating the cylinder-head temperatures with the primary engine and coolant variables and this method was successfully applied to the data.

Following the investigation on the single-cylinder engine (reference 1), a comprehensive investigation of the cooling characteristics of a multicylinder engine of 1710-cubic-inch displacement was conducted. The primary data obtained in this investigation are presented in reference 3, which presents plots of the cylinder temperatures and the coolant heat rejection against the basic engine and the coolant variables. In order to determine the applicability of the correlation method of reference 1 to a multicylinder engine, this method was employed in slightly modified form to correlate the cylinder-head temperature data of reference 3, and the results are presented herein. In reference 4, a variation of this correlation method is used to correlate the coolant heat rejection data of reference 3 with the basic engine and coolant parameters.

The data used in the correlation presented in this report cover wide ranges of engine and coolant conditions including engine power outputs up to 1860 brake horsepower, coolant flows from 50 to 320 gallons per minute, and coolants composed of ethylene glycol - water mixtures ranging in composition from 100-percent water to 97-percent ethylene glycol and 3-percent water.

APPARATUS AND PROCEDURE

The investigation of reference 3, which provided the data for this analysis, was conducted on a dynamometer stand. The data used in the analysis were obtained from four V-1710 engines

which are designated engines A, B, C, and D in reference 3 and herein. All these engines are standard production models; engine B was equipped with an aftercooler for part of the investigation. These engine models are 12-cylinder, liquid-cooled, V-type engines with a displacement of 1710 cubic inches, a 5.5-inch bore, and a 6.0-inch stroke. The compression ratio is 6.65 and the engines are fitted with single-stage gear-driven superchargers having a gear ratio of 9.6:1 and an impeller diameter of 9.5 inches. The standard ignition system is timed to fire the intake spark plugs 28° B.T.C. and the exhaust spark plugs 34° B.T.C. The valve overlap extends over a period of time equivalent to 74° rotation of the crankshaft.

The cylinder-head temperatures were measured by iron-constantan thermocouples installed in each cylinder between the exhaust valves, between the intake valves, and in the exhaust spark-plug boss at the locations shown in figure 1. A schematic diagram of the coolant system is shown in figure 2; further details of the instrumentation and a description of the general setup and auxiliary equipment are given in reference 3.

A summary of the engine and coolant conditions covered by the investigation is presented in table I. The tests on engine A covered typical engine operating conditions varying from cruise to take-off power and included data for several coolant flows and temperatures and a range of engine coolant-outlet pressures from 10 to 30 pounds per square inch gage. The effects of coolant temperature and engine power on the cylinder temperatures and coolant heat rejection were further determined in the part of the investigation conducted on engine B. Tests in which an aftercooler was mounted on this engine were also made to determine the effects of varying the charge flow, the manifold temperature, the fuel-air ratio, and the aftercooling conditions on engine cooling. The part of the investigation on engines C and D was conducted to extend the range of this correlation and to provide data essential to the correlation that were not obtained on the other two engines.

CORRELATION METHODS

Symbols

The following symbols are used in the correlation:

$B_1 \dots B_5$ constants

c specific heat of coolant, (Btu)/(lb)($^{\circ}$ F)

c_p	specific heat of air at constant pressure, (Btu)/(lb)(°F)
g	acceleration due to gravity, 32.2 (ft)/(sec ²)
J	mechanical equivalent of heat, 778 (ft-lb)/(Btu)
k	thermal conductivity of coolant, (Btu)/(sec)(sq ft)(°F/ft)
m, n, s	exponents
N	engine speed, (rpm)
Pr	Prandtl number of coolant, $\frac{c_p \mu}{k}$
T_c	carburetor inlet-air temperature, (°F)
T_g	effective cylinder-gas temperature, (°F)
T_h	average cylinder-head temperature, (°F)
T_l	average coolant temperature, (°F)
T_m	dry inlet-manifold temperature, (°F)
U	supercharger impeller tip speed, (ft)/(sec)
W_c	engine charge flow (air plus fuel), (lb)/(sec)
W_l	coolant flow, (lb)/(sec)
Z	factor that accounts for temperature drop through cylinder head
μ	absolute viscosity of coolant, (lb)/(ft)(sec)

Correlation Equation

One form of the equation developed in reference 1 for correlating the cylinder temperature with the primary engine and coolant conditions is

$$\left[\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c^n} \right) - Z \right] k Pr^s = B_1 \left(\frac{W_l}{\mu} \right)^{-m} \quad (1)$$

where T_g is a function of fuel-air ratio, inlet-manifold temperature, ignition timing, and exhaust pressure, k , μ , and Pr are functions of the coolant composition and temperature, and Z is a factor that accounts for the temperature drop through the cylinder head.

In equation (1), the coolant-flow factor W_l/μ is the separated variable, the effect of charge flow being incorporated with the temperature parameter. It may be desirable in many cases, however, to separate out the effect of charge flow. By rearrangement of equation (1), the following alternative form is obtained in which the charge flow is the separated variable:

$$\left(\frac{T_g - T_h}{T_h - T_l} \right) \left[\left(\frac{B_1}{W_l^m} \right) \left(\frac{\mu^m}{k Pr^s} \right) + Z \right] = W_c^{-n} \quad (2)$$

The constant B_1 , the factor Z , and the exponents m , n , and s are determined from the test results and the details of their evaluation are given later in this report. The significance of other factors appearing in the correlation equation or used in the analysis is discussed in the following section. When these factors are evaluated, the relations among T_h and the various operating conditions will be completely defined by equation (1) or (2), and these equations will then serve to correlate and permit the prediction of the cylinder-head temperature for any engine and coolant operating conditions.

If modes of heat transfer other than normal forced convection are predominant or if variables other than those contained in the correlation equation have an effect on the cylinder temperatures, the data may be expected to depart from a satisfactory correlation. Two such factors that may be encountered in engine operation are boiling of the coolant and scale build-up on the coolant passages. The effects of these two factors are illustrated in figures to be presented subsequently.

Significance of Factors

Cylinder temperature T_h . - The average of the temperatures measured between the exhaust valves for the 12 cylinders is used for the cylinder temperature T_h in this analysis. This temperature location was chosen because it is the most widely used temperature for this engine and, being in the hottest region of the cylinder head, may be considered indicative of critical cooling conditions. Although an average cylinder-head temperature is indicated in the derivation of the equation and is used in the correlation presented in reference 1 a satisfactory correlation would be expected for any single temperature location. This possibility of satisfactorily correlating the temperature of any location is a result of the linear relation (reference 3) that exists between temperatures at various locations in the cylinder head.

In order to permit an evaluation of the maximum cylinder-head temperature obtained for any operating condition, the relation between the average temperature for the 12 cylinders and the temperature of the hottest cylinder is presented.

Effective cylinder-gas temperature T_g . - The effective cylinder-gas temperature T_g is the gas temperature effective in transferring heat from the cylinder gases to the cylinders, and, as previously indicated, is considered a function of the fuel-air ratio, inlet-manifold temperature, ignition timing, and exhaust pressure. The value of T_g for the various engine conditions is determined from tests described and illustrated subsequently.

Dry inlet-manifold temperature T_m . - The true inlet-manifold temperature in a conventional multicylinder engine is difficult to measure because of the presence of unevaporated fuel in the charge mixture. For cooling correlations, the expedient of using a calculated dry inlet-manifold temperature instead of the measured manifold temperature is adopted. This dry inlet-manifold temperature is defined as the sum of the air temperature at the carburetor inlet and the calculated temperature rise of the air incurred in passing through the supercharger. This temperature rise was calculated on the assumption that there was no fuel vaporization. By assuming a value of 0.96 for the slip factor, which is the ratio of the pressure coefficient to the adiabatic efficiency of the supercharger, the dry inlet-manifold temperature T_m may be written as

$$T_m = T_c + \frac{0.96 U^2}{J g c_p} \quad (3)$$

For the single-stage engines used, this relation reduces to

$$T_m = T_c + 25.28 \left(\frac{N}{1000} \right)^2 \quad (4)$$

For the tests of the engine fitted with the aftercooler, the temperature drop of the charge mixture incurred in passing through the aftercooler was calculated from the heat rejected to the aftercooler coolant and subtracted from the manifold temperature determined from equation (4).

Charge flow W_c and fuel-air ratio. - The value of charge flow W_c was taken as the total charge flow (air plus fuel) to the engine, although similar correlations may also be made on the basis of the air flow alone. The fuel-air ratio used was the mean fuel-air ratio to all cylinders as obtained from the total air and fuel flows.

Coolant flow W_l and coolant temperature T_l . - The coolant flow W_l was, for simplicity, taken as the total coolant flow to both cylinder banks. Although the construction of the coolant passages in the engine is such that the flow varies considerably from cylinder to cylinder, it is shown in reference 3 that changes in the total flow effect proportional changes in the flow over any one cylinder. The coolant temperature T_l was taken as the average of the inlet and outlet temperatures of both cylinder banks.

Physical properties of coolants. - The physical properties of the coolants (specific heat c , absolute viscosity μ , thermal conductivity k , and therefore the Prandtl number Pr) were evaluated at the average coolant temperature T_l . The values used are presented in convenient curve form in reference 1.

EVALUATION OF FACTORS

Effective Cylinder-Gas Temperature T_g

The method used to evaluate T_g , which has been successfully applied to the correlation of cooling data obtained for a large

number of air-cooled engines and the liquid-cooled engine of reference 1, constitutes first the establishment of a reference value of T_g for a given set of operating conditions and then the determination of the variation of T_g with each of the pertinent engine conditions. On the basis of previous correlation work, a reference value of 1150° F for T_g was chosen for a fuel-air ratio of 0.080, an inlet-manifold temperature of 80° F, standard ignition timing (approximately maximum power setting), and an exhaust pressure of 30 inches of mercury absolute. Investigation has shown that the correlation is insensitive to changes in the magnitude of this reference value of T_g ; it is important, however, that its variation with engine conditions be accurately determined.

The variation of T_g with fuel-air ratio, manifold temperature, ignition timing, and exhaust pressure was determined from the tests in which these factors were independently varied while holding all other engine and coolant conditions constant. For such conditions, correlation equation (1) reduces to

$$\frac{T_h - T_l}{T_g - T_h} = \text{constant} \quad (5)$$

This constant is evaluated from the cylinder-head and coolant-temperature data at the reference operating conditions for which the value of T_g has already been established. The value of T_g for other than the reference value of the aforementioned variables is then calculated from the value of the constant and the coolant and cylinder-head temperatures obtained at the operating conditions in question. The engine and coolant conditions for which each of these tests was conducted are listed under each pertinent variable in table I.

Variation of T_g with fuel-air ratio. - The variation of T_g with fuel-air ratio is shown in figure 3. The values of T_g presented have been corrected to a dry inlet-manifold temperature of 80° F in accordance with a relation between the manifold temperature and T_g that will be subsequently discussed. A maximum value of T_g is reached at a fuel-air ratio of about 0.067, which is approximately equal to the fuel-air ratio for the stoichiometric mixture. Data obtained from three different engines, one of which was fitted with an aftercooler, are included in figure 3 and close agreement among the three is noted. The variations obtained for both the single-cylinder engine of reference 1 and the air-cooled

engine of reference 5 are also shown in this figure. The values of T_g for the single-cylinder engine are seen to be somewhat lower than those for the multicylinder engines of the present investigation, particularly in the rich region, but close agreement between the multicylinder liquid-cooled and air-cooled engines is evident.

Variation of T_g with dry inlet-manifold temperature T_m . - The variation of T_g with the calculated dry inlet-manifold temperature T_m is shown in figure 4. The data, which are presented for three engines and various engine operating conditions, have been adjusted to a fuel-air ratio of 0.080 in accordance with the relation between T_g and the fuel-air ratio that is presented in figure 3. As indicated in equation (4), the inlet-manifold temperature may be varied by changing either the carburetor inlet-air temperature or the engine speed. Data obtained from tests wherein each of these quantities was independently varied are presented (fig. 4) and both sets of data fall on a common curve.

An average change in T_g of about 0.25°F per degree Fahrenheit change in T_m is indicated and correction of T_g to other than 80°F manifold temperature is therefore made in accordance with the relation

$$\Delta T_g = 0.25 (T_m - 80) \quad (6)$$

Variation of T_g with exhaust pressure. - The effect of exhaust pressure on T_g for three values of fuel-air ratio and various engine conditions is presented in figure 5. These values of T_g have been corrected to an inlet-manifold temperature of 80°F by means of equation (6). An increase in exhaust pressure results in an increase in T_g , and, for the range of fuel-air ratios covered, the increase is somewhat greater at the lean than at the rich mixtures. A similar effect of exhaust pressure on T_g was obtained in the tests of an air-cooled engine which are reported in reference 6 and also in unpublished tests of another liquid-cooled engine conducted at this laboratory.

For convenience, a cross plot of these curves is shown in figure 6 in which T_g is plotted as a function of fuel-air ratio for exhaust pressures of 10, 20, 30, 40, and 50 inches of mercury

absolute. The curve for an exhaust pressure of 30 inches of mercury was obtained from figure 3 and the shape of this curve was used as a guide in drawing those for the other exhaust pressures.

Variation of T_g with ignition timing. - The variation of T_g with ignition timing for engine speeds of 2600 and 3000 rpm is presented in figure 7 as a plot of ΔT_g against spark advance. This variable ΔT_g represents the change in effective cylinder-gas temperature from the value at the standard ignition timing (exhaust spark plugs) of 34° B.T.C. The magnitude of the correction to T_g for other than standard ignition timing increases positively as the spark setting is increased or decreased from the normal position. The data separate somewhat with engine speed for spark advances less than the normal setting but, for simplicity, a single curve has been drawn through all the data.

Exponent n on Charge Flow

The value of the exponent n on charge flow W_c was obtained from tests at constant coolant conditions in which the charge flow was varied by changing the engine speed, manifold pressure, or exhaust pressure. For such conditions, equation (1) reduces to

$$\frac{T_h - T_l}{T_g - T_h} = B_2 W_c^n \quad (7)$$

A logarithmic plot of $\frac{T_h - T_l}{T_g - T_h}$ against W_c is shown in figure 8 and the slope of the line through the data, which is equal to the value of the exponent n , is 0.60. Although the absolute value of the factor $\frac{T_h - T_l}{T_g - T_h}$ would be different for different coolant conditions, the slopes of the resulting lines would be the same. The data for the runs with a coolant flow of 300 gallons per minute were adjusted to a flow of 250 gallons per minute to be consistent with the rest of the data by changing the factor $\frac{T_h - T_l}{T_g - T_h}$ in accordance with the effect of coolant flow on the cylinder-head temperature presented in reference 3. Data covering a wide range of engine speed, manifold pressure, and exhaust pressure are presented for each of two engines and close agreement is noted.

The Factor Z

The factor Z was determined from tests in which the coolant temperature and composition were held constant while the coolant flow W_l was varied. For these conditions, equation (1) may be written

$$\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c n} \right) - Z = B_3 \left(\frac{1}{W_l} \right)^m \quad (8)$$

A plot of $\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c 0.60} \right)$ against $1/W_l$, using the previously determined values of T_g and the exponent n , is shown in figure 9 for five different coolant compositions. Extrapolation of these data to $1/W_l = 0$ (at which point the value of $\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c 0.60} \right)$ is equal to the Z factor) results in a value of 0.13 for Z. Inasmuch as the extrapolation was necessarily made over a fairly wide range of flow rates, it may be considered somewhat arbitrary. The value of 0.13 was chosen by drawing curves having the same general shape as those presented for a similar plot in reference 1 and then plotting a final correlation using three different values of Z; the value that gave the most satisfactory correlation (0.13) was finally chosen.

In the investigation of reference 3, it was found that the cylinder-head temperature, particularly in the exhaust or hot side of the head, increased with engine running time during the initial operation of the engine. This increase in temperature is illustrated in figure 10 and it is noted that the temperature between the exhaust valves increased about 25° F during approximately the initial 100 hours of engine running time and remained substantially constant as the operating time was further increased. Unlike the variation exhibited by this temperature location, the temperature at the exhaust spark-plug boss increased only slightly and the temperature between the intake valves remained constant over the entire period of the investigation. As discussed in reference 3, an inspection of the coolant passages of a scrapped cylinder head revealed scale deposits on the exhaust side of the cylinder head but none on the intake side. This increase in temperature was therefore attributed to the scale deposits on the coolant passages. Because the Z factor accounts for the temperature drop through the cylinder head, this increase in cylinder-head temperature with engine running time will be reflected as a similar variation in

the Z factor. The determination of the Z factor (0.13), which was made after about 100 hours accumulated engine running time, is therefore applicable only for this or greater engine running times. Although the final correlation is insensitive to the magnitude of this basic value of Z, it is necessary that the variation of Z, which occurs during the initial engine running time, be accurately determined and included in the final correlation.

The variation of Z during the initial engine running time was determined from cylinder-head-temperature data obtained at a reference operating condition. For this reference condition, the coolant and engine conditions were constant; accordingly, equation (8) may be written

$$\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c^{0.60}} \right) - Z = \text{constant} \quad (9)$$

The value of the constant is determined from the substitution into the equation of the previously determined value of Z and the pertinent engine and coolant data obtained at 100 hours engine running time. The value of Z at other engine running times is then determined from the value of the constant and the cylinder-head temperatures obtained at the running time under consideration.

A plot showing the variation of the Z factor for engine D with engine running time is presented in figure 11. Some of the data were obtained at the reference operating condition; other data, included in order to establish more completely the variation of Z with time, were obtained at engine conditions other than the reference condition and adjusted in accordance with the variation in T_g and the effect of W_c previously presented. It can be seen that the Z factor increases during the initial engine operating time in a manner similar to that for the cylinder-head temperature (fig. 10) and that there is no significant change in the value of Z after about 100 hours running time. It is expected that this phenomenon will vary from engine to engine depending upon the history of operation. The range of conditions and engine running time over which this variation was established is indicated by the schedule of engine operating time in table I.

For engine D, the value of Z used in the subsequent correlation plots was determined from the curve of figure 11. For engines A, B, and C, the data were insufficient to provide separate evaluation of Z, but most of the correlation data provided by these engines were obtained after the engines had been run for a

considerable length of time so a value of 0.13 was used. Because this value resulted in satisfactory correlation of the data from engines A, B, and C with those of engine D, it may be assumed that the coolant passages of these engines were in about the same condition as those of engine D after about 100 hours engine running time.

Exponent m on Coolant-Flow Parameter W_l/μ

The value of the exponent m on the coolant-flow parameter W_l/μ was determined from tests in which the coolant temperature and composition were held constant while the coolant flow W_l was varied (similar to data for determination of Z factor). For these conditions, equation (1) may be written

$$\left[\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c^n} \right) - Z \right] k = B_4 \left(\frac{W_l}{\mu} \right)^{-m} \quad (10)$$

A logarithmic plot of the factor $\left[\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c^{0.60}} \right) - Z \right] k$ against W_l/μ is shown in figure 12 for five different coolant compositions. The slope of the straight lines through these data is equal to the exponent on W_l/μ . Lines having the same slope are drawn through the data for each coolant and the value of the exponent m is accordingly established as 0.48. This value for the exponent m , which is established from these selected data, will, of course, be verified by the slope of the line through all the data on the final correlation plot based on equation (1).

Exponent s on Prandtl Number Pr

The value of the exponent s on the Prandtl number Pr was determined from data for a constant value of W_l/μ and for constant engine conditions. For these conditions, equation (1) may be reduced to

$$\left[\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c^n} \right) - Z \right] k = B_5 Pr^{-s} \quad (11)$$

The slope of the line determined by a logarithmic plot of

$$\left[\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c 0.60} \right) - Z \right] k \text{ against } Pr \text{ (or } \frac{c\mu}{k} \text{) would then}$$

establish the value of the exponent s . A wide range of Prandtl number for this plot may be obtained if data for several different coolants are used. In order to construct this plot, it is convenient to obtain values of the factor

$$\left[\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c 0.60} \right) - Z \right] k$$

from figure 12 for a constant value of W_l/μ and then to cross-plot the values of this factor against the Prandtl number Pr of the different coolants. Data obtained from a cross plot of figure 12 at a constant value of W_l/μ equal to 55,000 is shown in figure 13 and the slope of this line thus establishes the value of the exponent s on the Prandtl number Pr as 0.33.

FINAL CORRELATION

Final Correlation with Coolant-Flow Factor W_l/μ as

Independent Variable

The final correlation based on equation (1), which is obtained by plotting the factor

$$\left[\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c 0.60} \right) - Z \right] k Pr^{0.33}$$

against W_l/μ for all test data on logarithmic coordinates, is presented in figure 14(a). Although the data points scatter considerably, the maximum variation in T_h resulting from this scatter is not large. A variation in T_h of $\pm 12^\circ F$ for typical engine conditions at normal rated power is represented by the dashed lines in the figure, and it is seen that almost all the data for the four engines lie within this band.

The value of the exponent m , which is obtained from the slope of the line through the data, is, as previously determined, equal to 0.48 and the value of the constant B_1 , found by substitution of the values of the coordinate of any point on the line into the correlation equation, is equal to 0.00163. The final equation is accordingly written

$$\left[\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c^{0.60}} \right) - Z \right] k Pr^{0.33} = 0.00163 \left(\frac{W_l}{\mu} \right)^{-0.48} \quad (12)$$

and will apply over a considerable range of engine operating conditions, coolant temperatures, coolant flows, and coolant compositions.

In order to illustrate the variation in the correlation obtained among the four engines, the correlation of figure 14(a) is replotted in figure 14(b) using a different symbol for each engine. The separation of the data between engines is slight and the scatter for one engine is almost as great as the total scatter for all engines.

Final Correlation with Charge Flow W_c as Independent Variable

Correlation of the test data based on equation (2), wherein the factor

$$\left(\frac{T_g - T_h}{T_h - T_l} \right) \left[\left(\frac{0.00163}{W_l^{0.48}} \right) \left(\frac{\mu^{0.48}}{k Pr^{0.33}} \right) + Z \right]$$

is evaluated by using the previously determined values of m , s , Z , and B_1 and plotted against the charge flow W_c , is shown in figure 15. A straight line with a slope of -0.60 , which is equal in absolute value to the value of the exponent n on W_c , previously determined, is drawn through the data. As a result of the different arrangement of the terms in this equation, the apparent scatter of the data is considerably less in figure 15 than in figure 14; the over-all accuracy of the correlation is, of course, the same.

When equation (12) is rearranged in accordance with this plot, the following form is obtained:

$$\left(\frac{T_g - T_h}{T_h - T_l} \right) \left[\left(\frac{0.00163}{W_l^{0.48}} \right) \left(\frac{\mu^{0.48}}{k Pr^{0.33}} \right) + Z \right] = W_c^{-0.60} \quad (13)$$

As previously mentioned, the values of the factor Z for engine D used in plotting the final correlation of figures 14 and 15 were obtained from figure 11. If a constant average value were

used for Z , the scatter of the data would be increased from approximately $\pm 12^\circ$ to about $\pm 22^\circ$ F, depending upon the engine running time.

In order to facilitate the computation of head temperatures by means of equation (13), values of the coolant-property

parameter $\left(\frac{\mu^{0.48}}{k \text{ Pr}^{0.33}} \right)$ are presented in figure 16 for various

coolant mixtures of ethylene glycol and water over a range of coolant temperatures.

Effect of Boiling of Coolant on Correlation

It was found in the investigations of reference 3 that under some conditions of operation a reduction in coolant flow increased the amount of boiling of the coolant and reduced the normal tendency of the cylinder-head temperature to increase with reduced coolant flow. In order to determine the effect of this boiling of the coolant in the correlation, the relations given by equation (13) were considered. The left-hand side of this equation should normally be a function of only charge flow and, if boiling of the coolant were negligible, would be independent of the coolant flow. If, however, boiling occurs to an appreciable extent, the value of the left-hand side of this equation would be expected to vary with the coolant flow. A plot of this parameter against coolant flow is shown in figure 17 for two different engine operating conditions and several coolant compositions. For coolant flows greater than about 100 gallons per minute, the value of the plotted parameter is independent of the coolant flow, which indicates that boiling of the coolant did not occur to a noticeable degree in this range. For coolant flows less than about 100 gallons per minute, however, the value of the parameter increases with reduced coolant flow (depending on the engine power and coolant), which illustrates the tendency of boiling of the coolant in this range of flow rates to reduce the cylinder-head temperature.

The maximum variation of the parameter plotted in figure 17 is equivalent to a decrease in head temperature of less than 10° F. Because this variation is within the normal scatter of the data, the correlation is not seriously affected by boiling for the ranges of variables covered. Extrapolation of this correlation to combinations of higher engine power, lower coolant flows, or lower coolant pressures than those covered by this investigation would, however, be subject to uncertainties and reduced accuracy of prediction of cylinder temperatures.

Relation between Average and Maximum Cylinder-Head Temperatures

In figure 18 the average temperature of the 12 cylinders is plotted against the temperature of the hottest cylinder for the location in the cylinder head between exhaust valves. Good correlation of these data is obtained for all conditions and for all engines with the maximum cylinder-head temperature ranging from 10° to 20° F higher than the average temperature. From the correlation of the average cylinder-head temperature with the primary engine variables and coolant variables given in figure 14 or 15 and from the relation between the maximum and average head temperatures shown in figure 18, an estimation of the maximum cylinder-head temperature between exhaust valves is possible for a large range of engine and coolant conditions.

USE OF CORRELATION EQUATION

The use of the correlation equation in determining cylinder-head temperatures is illustrated by the following example:

The maximum cylinder-head temperature between the exhaust valves is to be determined for the following engine and coolant conditions:

Engine charge flow (air plus fuel), lb/sec	3.0
Engine speed, rpm	3000
Fuel-air ratio	0.095
Carburetor-inlet air temperature, $^{\circ}$ F	60
Exhaust pressure, in. Hg absolute	40.0
Ignition timing (exhaust spark plugs), deg B.T.C.	40
Accumulated engine running time, hr	Over 100
Coolant flow, lb/sec	30.0
Average coolant temperature, $^{\circ}$ F	250
Coolant composition, ethylene glycol - water (percent by volume)	30 - 70

The value of the average cylinder-head temperature is first evaluated from equation (13) and the maximum cylinder-head temperature then determined from figure 18. Although either equation (12) or (13) may be used to evaluate the average cylinder-head temperature, the grouping of the coolant-property terms of equation (13) results in greater convenience of application.

The dry inlet-manifold temperature is computed from the carburetor-air temperature and the engine speed by means of equation (4) as follows:

$$\begin{aligned}
 T_m &= T_c + 25.28 \left(\frac{N}{1000} \right)^2 \\
 &= 60 + 25.28 \left(\frac{3000}{1000} \right)^2 \\
 &= 288^\circ \text{ F}
 \end{aligned}$$

In order to determine the effective cylinder-gas temperature T_g , a value of T_g for a fuel-air ratio of 0.095, an exhaust pressure of 40 inches of mercury absolute, standard ignition timing, and a dry inlet-manifold temperature of 80° F is first determined from figure 6 as 1069° F . The correction for an ignition timing of 40° B.T.C. is obtained for figure 7 as 24° F . For a dry inlet-manifold temperature of 288° F , the correction to T_g is determined from equation (6)

$$\begin{aligned}
 \Delta T_g &= 0.25 (T_m - 80) \\
 &= 0.25 (288 - 80) \\
 &= 52^\circ \text{ F}
 \end{aligned}$$

The value of T_g is then determined by adding algebraically the corrections for ignition timing and manifold temperature to the value obtained from figure 6.

$$T_g = 1069 + 24 + 52 = 1145^\circ \text{ F}$$

Because the variation of Z with engine running time was shown to be constant at a value of 0.13 (fig. 11) for any engine running time over 100 hours, this constant value is used. The coolant-property parameter $\left(\frac{\mu^{0.48}}{k \text{ Pr}^{0.33}} \right)$ is determined from figure 16 for the specified coolant and coolant temperature as equal to 164.

Substitution of the values of the various parameters into equation (13)

$$\left(\frac{T_g - T_h}{T_h - T_l} \right) \left[\left(\frac{0.00163}{W_l^{0.48}} \right) \left(\frac{\mu^{0.48}}{k \text{ Pr}^{0.33}} \right) + Z \right] = W_c^{-0.60}$$

gives the following

$$\left(\frac{1145 - T_h}{T_h - 250} \right) \left[\left(\frac{0.00163}{30^{0.48}} \right) (164) + 0.13 \right] = 3.0^{-0.60}$$

$$\left(\frac{1145 - T_h}{T_h - 250} \right) (0.0522 + 0.13) = 0.5173$$

Therefore

$$\left(\frac{1145 - T_h}{T_h - 250} \right) = 2.839$$

Solving for the value of T_h ,

$$2.839 T_h + T_h = 1145 + 2.839 \times 250$$

$$3.839 T_h = 1855$$

$$T_h = 483^\circ \text{ F}$$

For this value of the average cylinder-head temperature, the maximum cylinder-head temperature between the exhaust valves is found to be 499° F (fig. 18).

SUMMARY OF RESULTS

An analysis of the data obtained from four multicylinder, liquid-cooled engines of 1710-cubic-inch displacement, which included power outputs up to 1860 brake horsepower, coolant flows from 50 to 320 gallons per minute, and coolants composed of ethylene glycol - water mixtures varying in composition from 100-percent water to 97-percent ethylene glycol and 3-percent water gave the following results:

1. The NACA correlation method, which is based on the theory of heat transfer by normal forced convection, provided satisfactory correlation of the cylinder-head temperature between the exhaust valves with the primary engine and coolant variables for a wide range of engine and coolant conditions.

2. The correlation method as applied herein permitted the prediction of the cylinder-head temperature between the exhaust valves within approximately $\pm 12^\circ \text{ F}$.

3. The increase in the cylinder-head temperatures, which occurred during about the first 100 hours of engine operating time for constant operating conditions, (attributed to scale build-up in the coolant passages) was incorporated in the correlation.

4. Boiling of the coolant, which was encountered at several engine powers under certain conditions of coolant flow, coolant temperature, and coolant composition, did not seriously affect the correlation for the ranges of variables covered in this investigation.

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2. Pinkel, Benjamin: Heat-Transfer Processes in Air-Cooled Engine Cylinders, NACA Rep. No. 612, 1938.
3. Povolny, John H., and Chelko, Louis J.: Cylinder-Head Temperatures and Coolant Heat Rejections of a Multicylinder, Liquid-Cooled Engine of 1710-Cubic-Inch Displacement. NACA TN No. 1606, 1948.
4. Povolny, John H., and Lundin, Bruce T.: Correlation of Coolant Heat Rejections of a Multicylinder, Liquid-Cooled Engine of 1710-Cubic-Inch Displacement. NACA RM No. E8B06a, 1948.
5. Manganiello, Eugene J., Valerino, Michael F., and Bell, E. Barton: High-Altitude Flight Cooling Investigation of a Radial Air-Cooled Engine. NACA TN No. 1089, 1946.
6. Valerino, Michael F., Kaufman, Samuel J., and Hughes, Richard F.: Effect of Exhaust Pressure on the Cooling Characteristics of an Air-Cooled Engine. NACA TN No. 1221, 1947.

TABLE I - SUMMARY OF ENGINE

Variable or type of test	Engine power (bhp)	Engine speed (rpm)	Manifold pressure (in. Hg absolute)	Charge flow (lb/sec)	Fuel-air ratio	Carburetor-air temperature (°F)
Charge flow	400-1180	2600	24-48	1.03-2.47	0.080	60
	275-1260	2600	21-54	.84-2.76	.095	85
	925-1860	3200	41-73	2.35-4.40	.095	-5
	860-940	2300-3200	40	1.70-2.18	.080	60
	750-873	1800-3200	40	1.54-2.14	.095	86
	430-630	1400-3200	30	.88-1.60	.095	86
	370-640	1200-3200	30	.73-1.57	.080	60
Manifold temperature	786-794	2600	34-36	1.83	0.095	80-184
	555-590	2600	27-30	1.39	.090	-20-170
	659-675	2600	30-34	1.58	.095	-27-144
	798-846	3000	36-40	2.04	.095	-43-146
	662-825	2300-3200	32-36	1.72	.080	17
	647-812	2300-3200	34-40	1.72	.080	140
	662-868	2000-3200	33-40	1.76	.095	17
	546-742	2000-3200	31-40	1.58	.095	143
Fuel-air ratio	740-790	2600	35	1.75	0.064-0.116	66
	731-837	2600	34-36	1.84	.062- .109	120
	760-960	3000	39-41	2.20	.064- .118	25
Ignition timing	577-656	2600	30	1.53	0.095	40
	780-922	3000	40	2.19	.095	19
Exhaust pressure	540-612	2600	28-35	1.43	0.063	24
	598-703	2600	30-40	1.60	.100	4
	620-726	2600	30-42	1.59	.085	39
	944-994	3000	40-47	2.28	.085	-2
	592-682	3000	30-41	1.66	.085	19
	427-882	2600	35	1.36-1.92	.085	32
Coolant flow	1000	2600	44	2.21	0.092	75
	1000	2600	44	2.21	.093	80
	780, 1160	2600, 3000	36, 50	1.78, 2.71	.095	83
	780, 1160	2600, 3000	36, 50	1.78, 2.71	.095	83
	780, 1160	2600, 3000	36, 50	1.78, 2.71	.095	83
	780, 1160	2600, 3000	36, 50	1.78, 2.71	.095	83
	780, 1160	2600, 3000	36, 50	1.78, 2.71	.095	83
Engine calibration	1000, 1250, 1450	3000	42, 51, 60	2.39, 2.94, 3.40	0.095	50
	310-1200	2000-3000	24-53	.73-2.80	.064-.096	75

^a Engine equipped with aftercooler.^b AN-E-2 ethylene glycol..

AND COOLANT CONDITIONS

Ignition timing (deg B.T.C.)		Exhaust pressure (in. Hg absolute)	Dry-inlet manifold temperature (°F)	Coolant, ethylene glycol-water (percent by volume)		Coolant flow (gal/min)	Average coolant temperature (°F)	Engine coolant-outlet pressure (lb/sq in. gage)	Engine	Engine running time (hr)
Exhaust	Intake			Glycol	Water					
34	28	29-30	230	30	70	255	245	33	C	-----
34	28		256	30	70	250	245	35	D	38.9- 45.8
34	28		254	30	70	300	245	35	D	153.8-156.5
34	28		194-319	30	70	255	245	33	C	-----
34	28		168-345	30	70	250	245	35	D	47.2- 51.2
34	28		135-345	30	70	250	245	35	D	45.8- 47.2
34	28		96-319	30	70	255	245	30	C	-----
34	28	29-30	223-275	30	70	230	245	25	B ^a	-----
34	28		150-340	30	70	270	245	30	C	-----
34	28		143-314	30	70	300	245	35	D	15.2- 19.7
34	28		165-374	30	70	300	245	35	D	19.7- 24.5
34	28		152-277	30	70	300	245	35	D	24.5- 28.3
34	28		274-396	30	70	300	245	35	D	28.3- 31.2
34	28		118-277	30	70	300	245	35	D	31.2- 33.7
34	28	29-30	244-403	30	70	300	245	35	D	33.7- 35.8
34	28		236	30	70	270	245	35	C	-----
34	28		229-250	30	70	220	245	27	B ^a	-----
34	28	29-30	252	30	70	250	245	35	D	37.5- 38.9
14-54 16-52	8-48 10-46		210 246	30 30	70 70	300 300	245 245	35 35	D D	112.2-116.1 116.1-119.3
34	28	6-45 10-60 10-61 12-52 12-59 10-61	194	30	70	250	245	35	D	148.5-151.4
34	28		174	30	70	250	245	35	D	142.2-145.8
34	28		209	30	70	250	245	35	D	136.5-138.5
34	28		226	30	70	250	245	35	D	140.6-142.2
34	28		246	30	70	250	245	35	D	138.5-140.6
34	28		202	30	70	250	245	35	D	145.8-148.5
34	28	29-30	245	97 ^b	3	200-300	215, 245, 270	10	A	-----
34	28		250	30	70	100-280	215, 245	10-30	A	-----
34	28		253, 310	0	100	50-300	245	35	D	85.6- 90.7
34	28		253, 310	30	70	50-300	245	35	D	90.7- 95.3
34	28		253, 310	50	50	50-300	245	35	D	95.3- 99.1
34	28		253, 310	70	30	50-300	245	35	D	99.1-103.1
34	28		253, 310	97 ^b	3	50-320	245	35	D	103.8-110.2
34	28	29-30	278	30	70	220	245, 270	20, 25, 35, 38	B	-----
34	28		176-303	97 ^b	3	144-255	245	10-30	A	-----



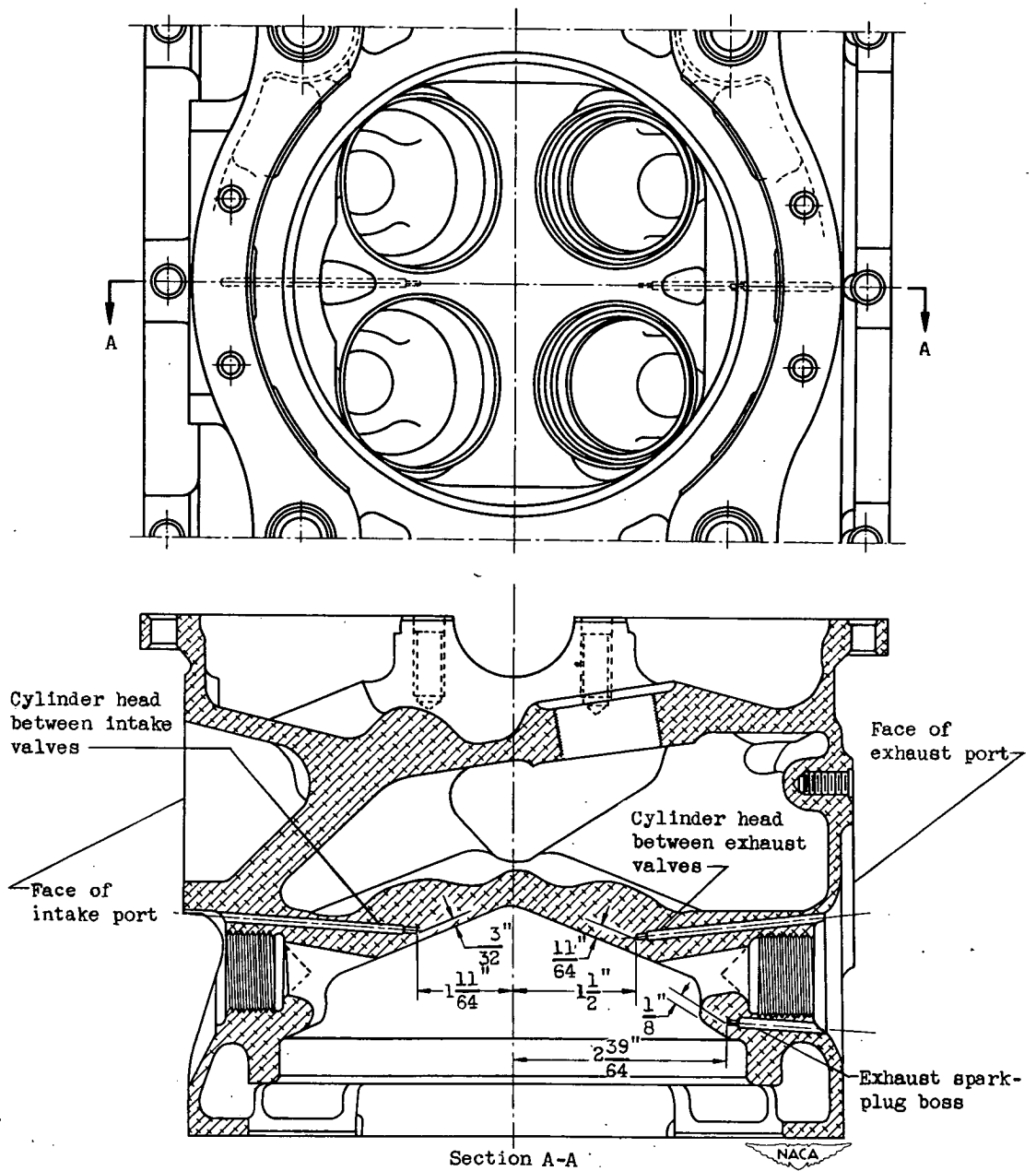


Figure 1. - Installation of cylinder-head thermocouples.

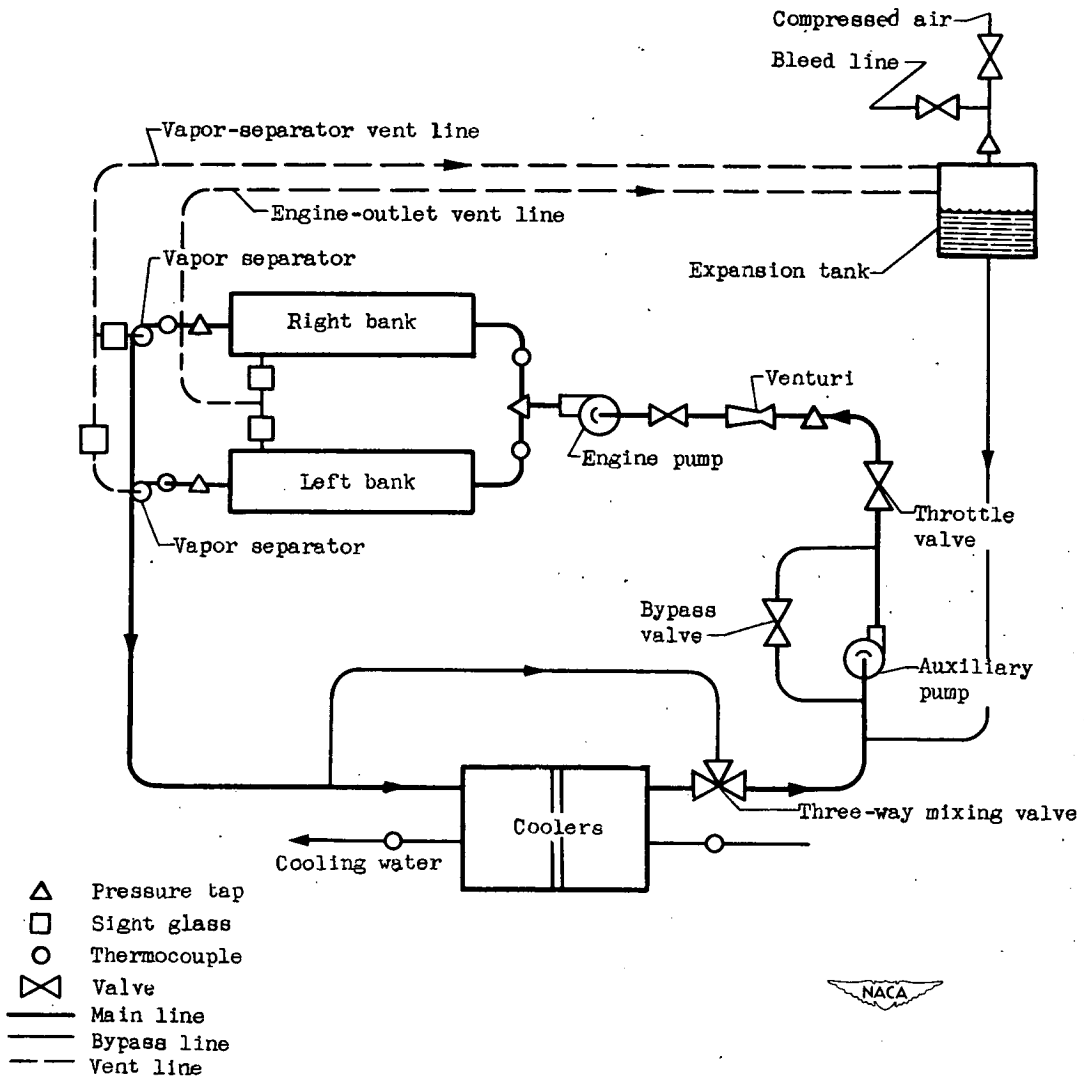


Figure 2. - Schematic diagram of engine coolant system.

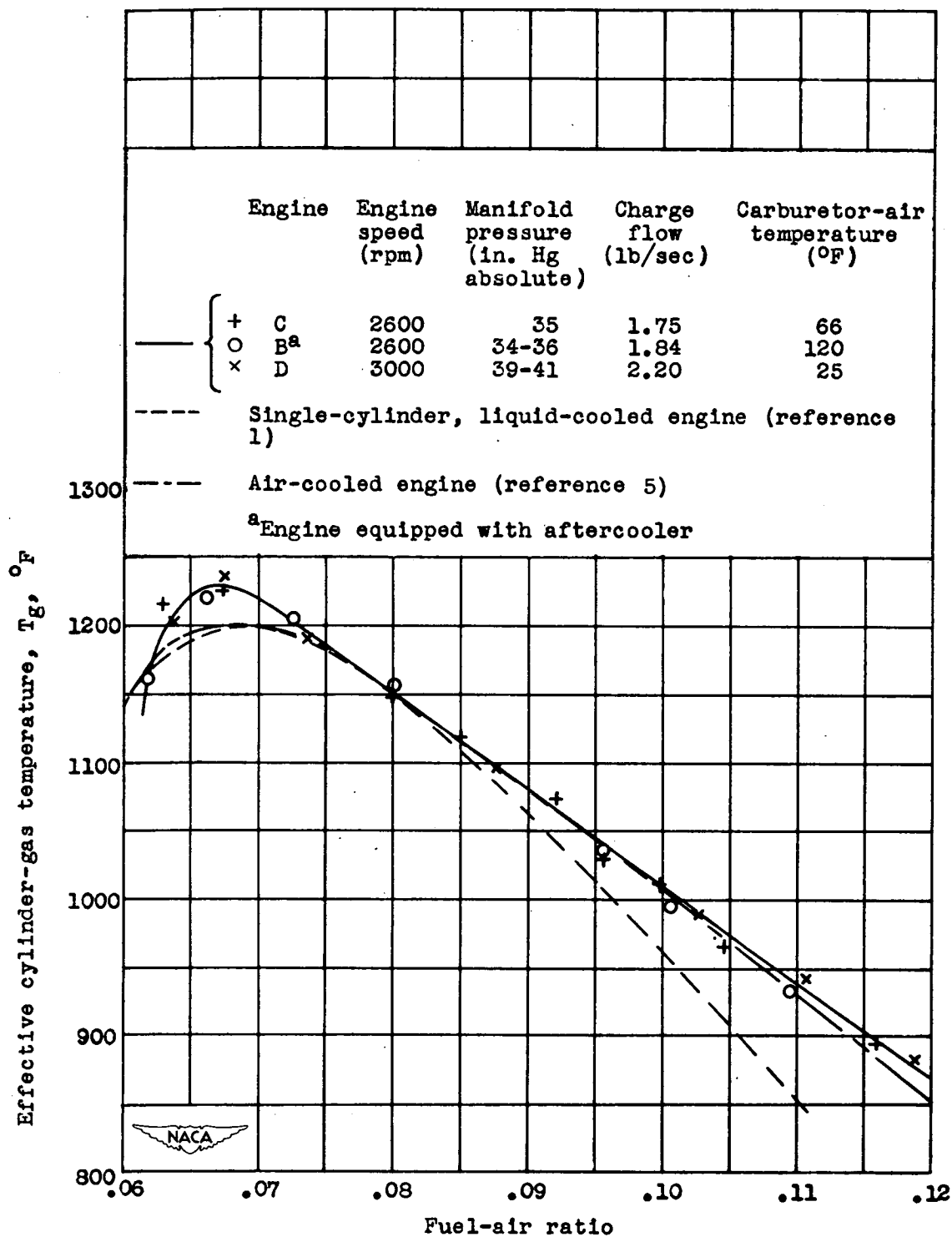


Figure 3. - Variation of effective cylinder-gas temperature with fuel-air ratio. Data corrected to dry inlet-manifold temperature of 80° F. Exhaust pressure, 29-30 inches mercury absolute; standard ignition timing.

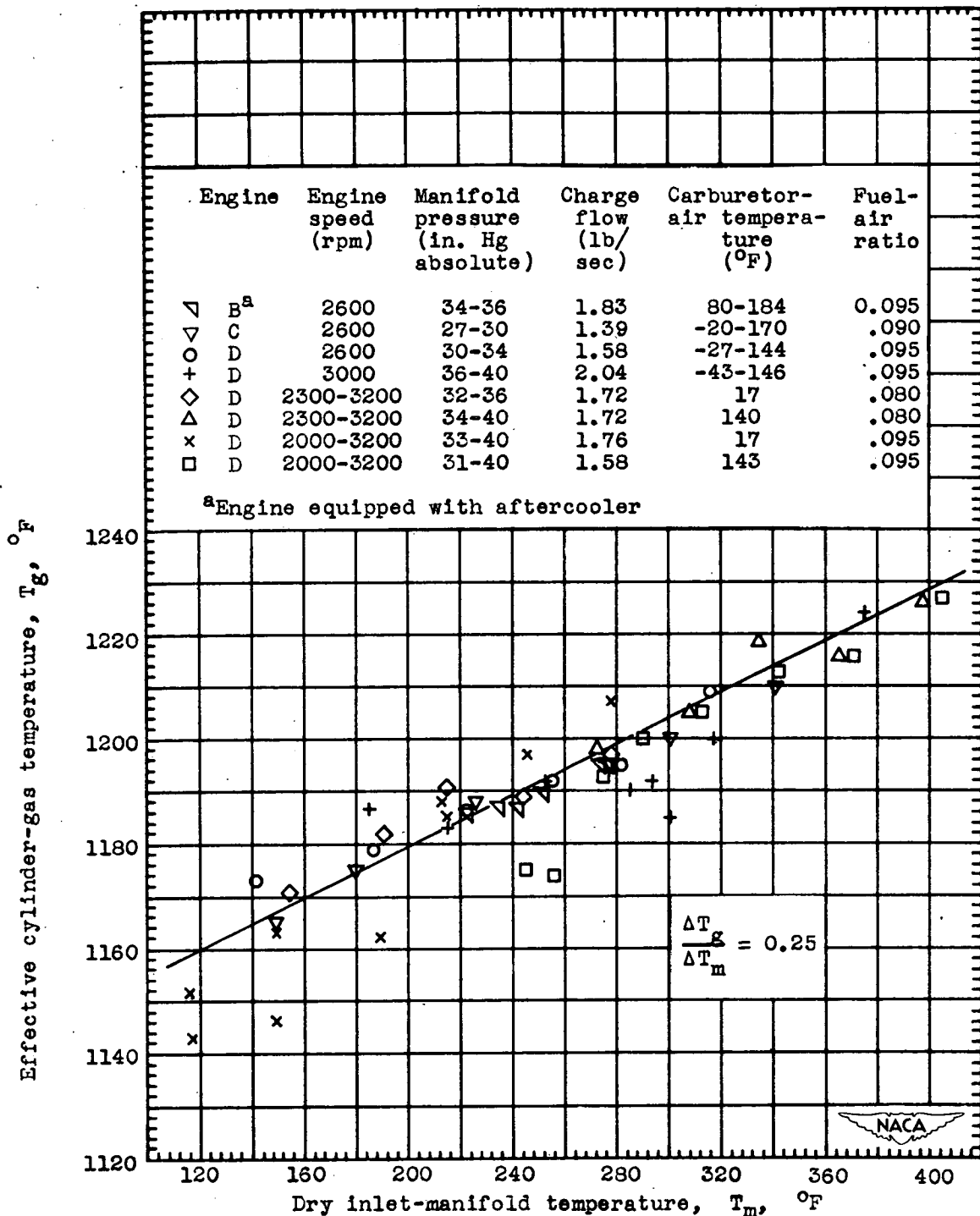


Figure 4. - Variation of effective cylinder-gas temperature with manifold temperature. Data corrected to fuel-air ratio of 0.080. Exhaust pressure, 29-30 inches mercury absolute; standard ignition timing.

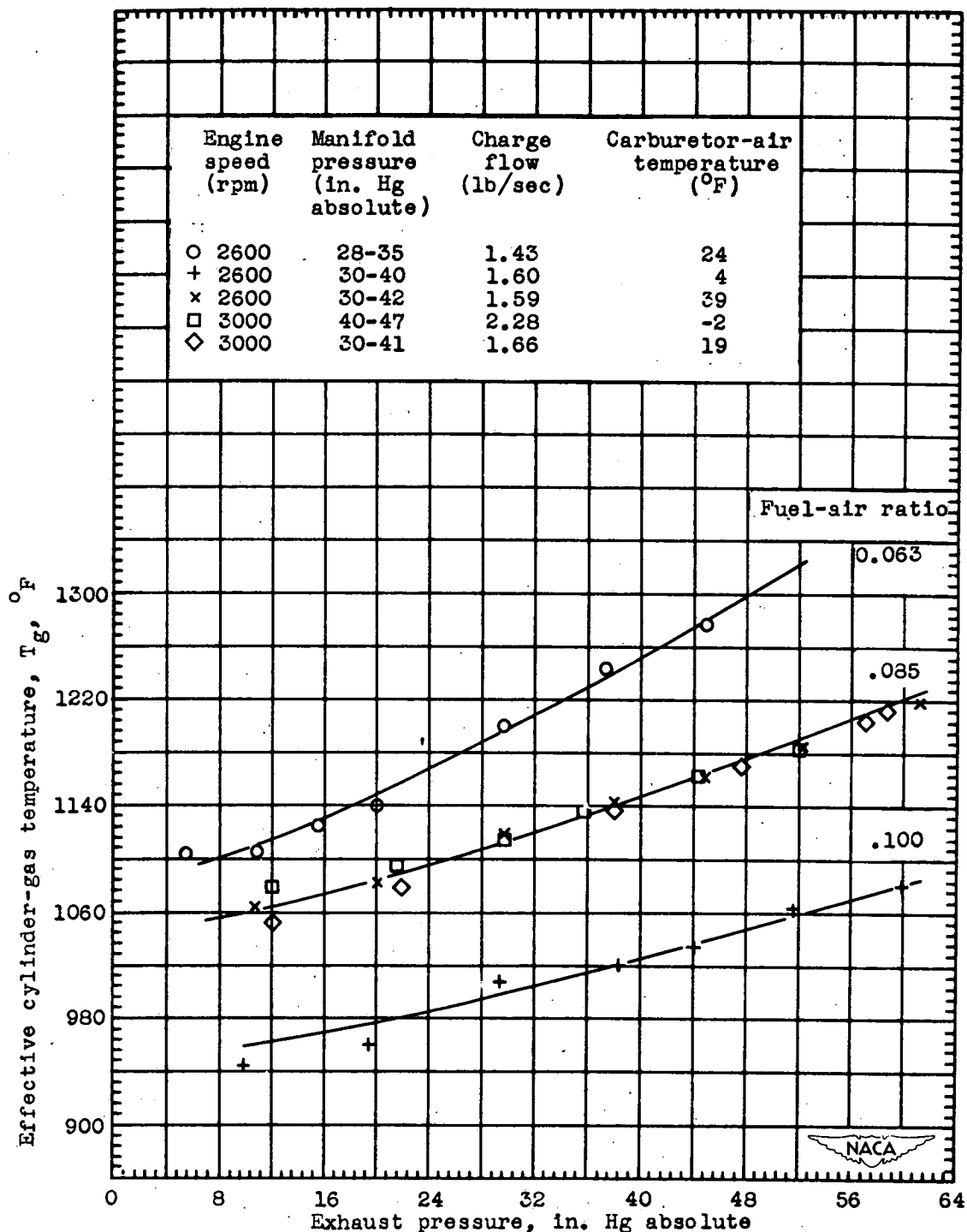


Figure 5. - Variation of effective cylinder-gas temperature with exhaust pressure at several fuel-air ratios. Data corrected to dry inlet-manifold temperature of 80° F; standard ignition timing; engine D.

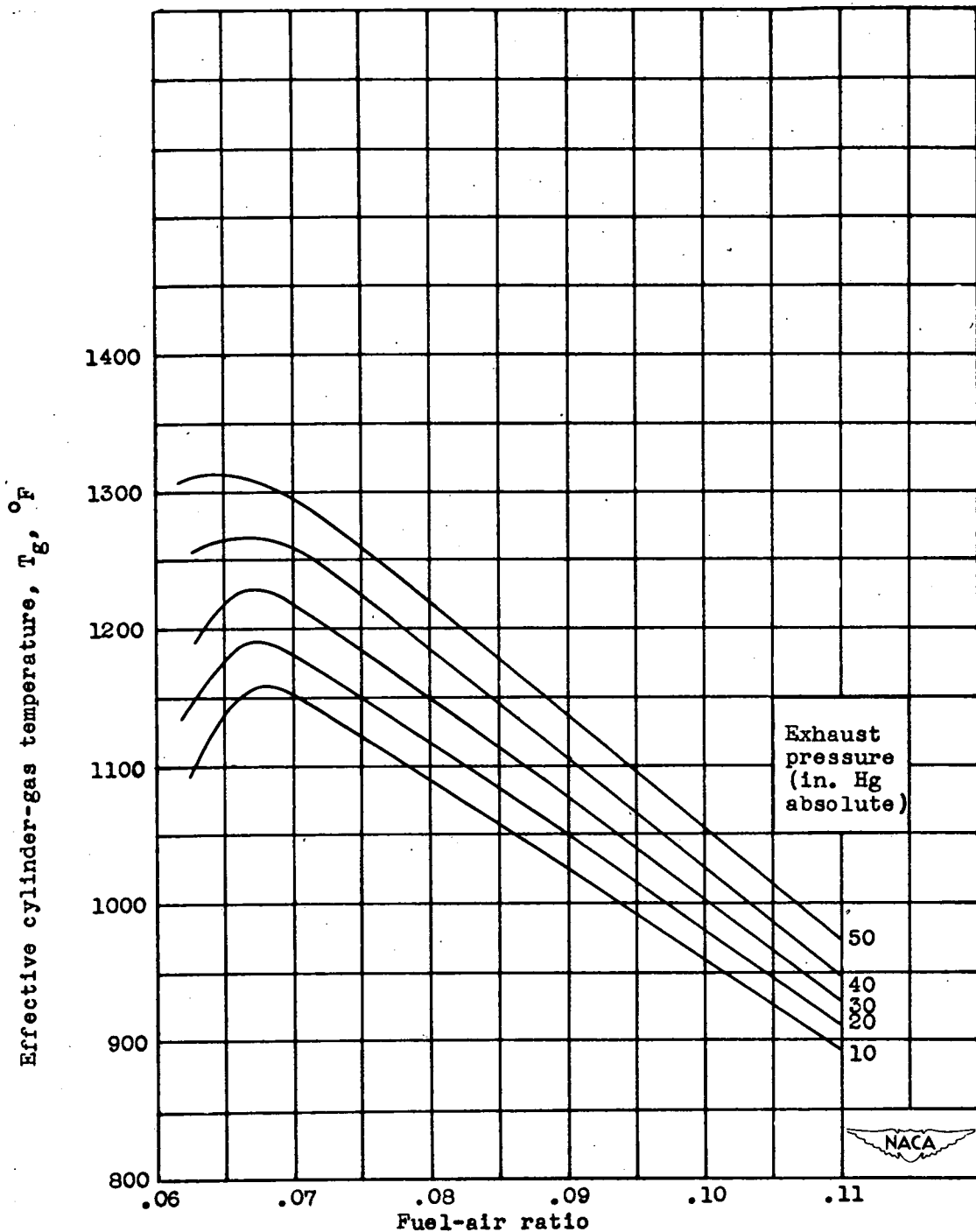


Figure 6. - Variation of effective cylinder-gas temperature with fuel-air ratio at various exhaust pressures obtained from cross plot of figure 5. Data corrected to dry inlet-manifold temperature of 80° F; standard ignition timing; engine D.

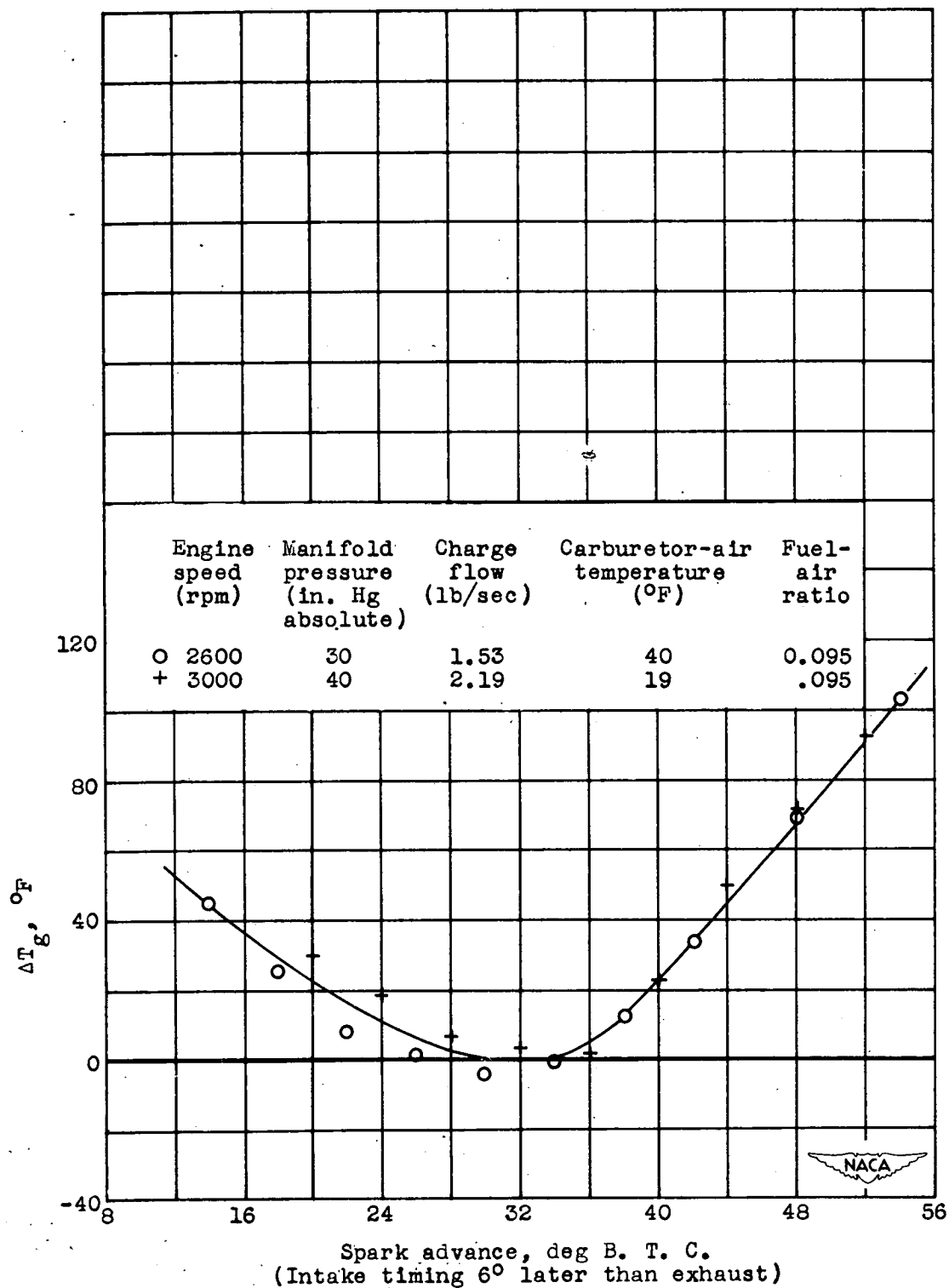


Figure 7. - Variation of change in effective cylinder-gas temperature with ignition timing. Engine D.

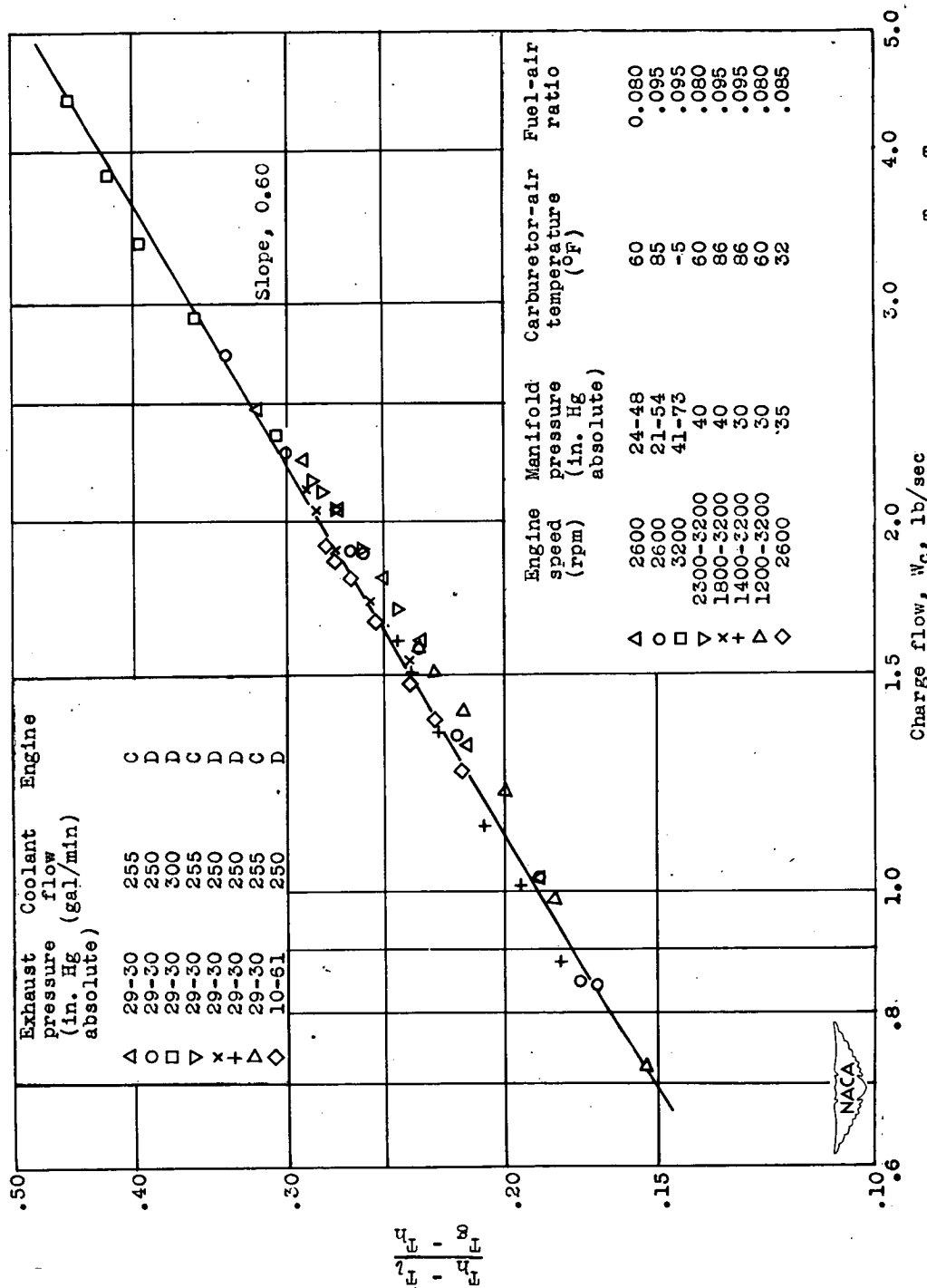


Figure 8. - Determination of exponent n on charge flow W_c from variation of $\frac{T_h - T_l}{T_g - T_h}$ with W_c . Coolant, 30-70 ethylene glycol - water; average coolant temperature, 245 $^{\circ}$ F.

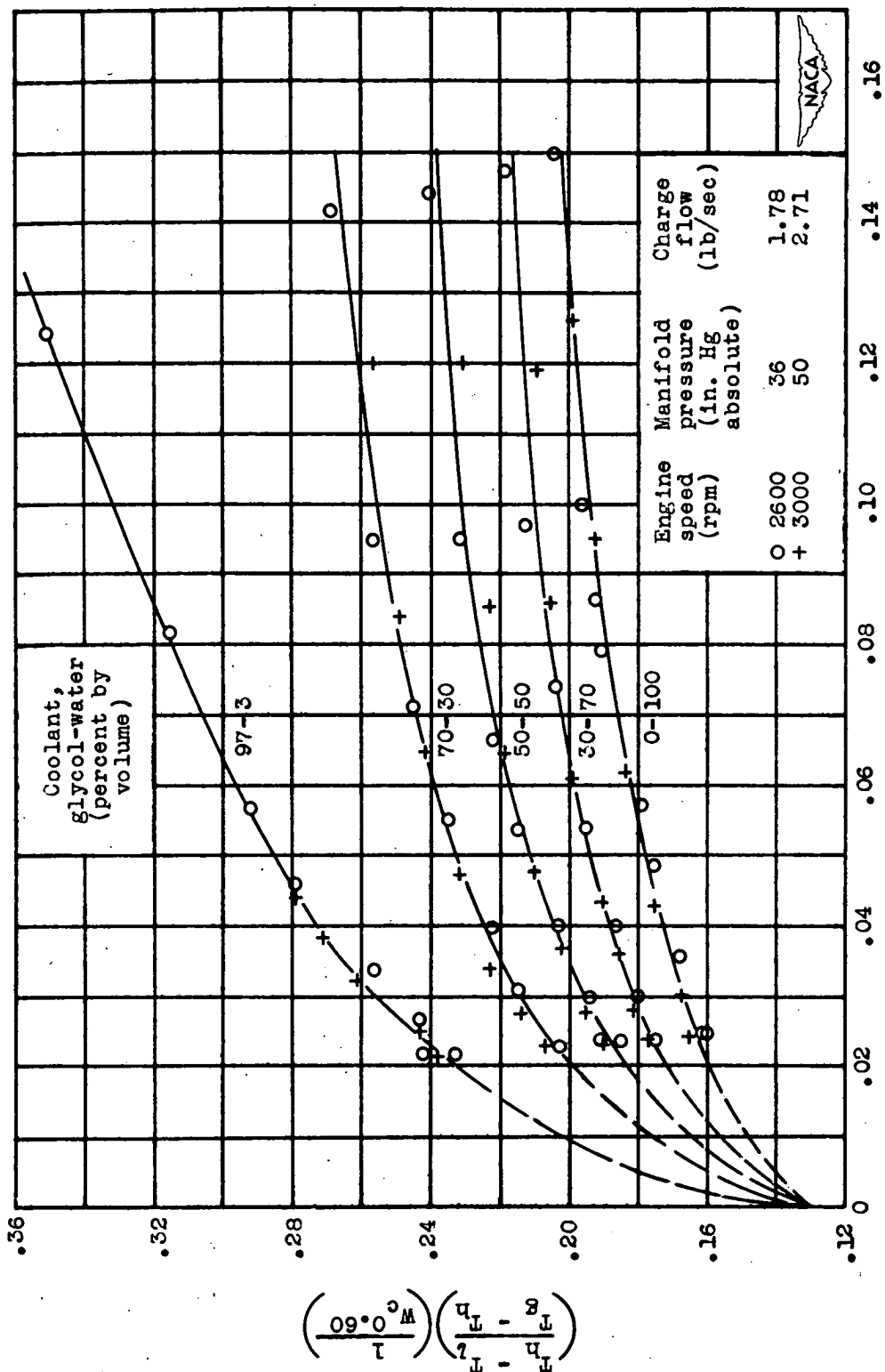


Figure 9. - Determination of factor Z from variation of $\left(\frac{T_h - T_l}{T_g - T_l} \right) \left(\frac{1}{W_c} \right)^{0.60}$ with l/W_l . Average coolant temperature, 245° F; engine D.

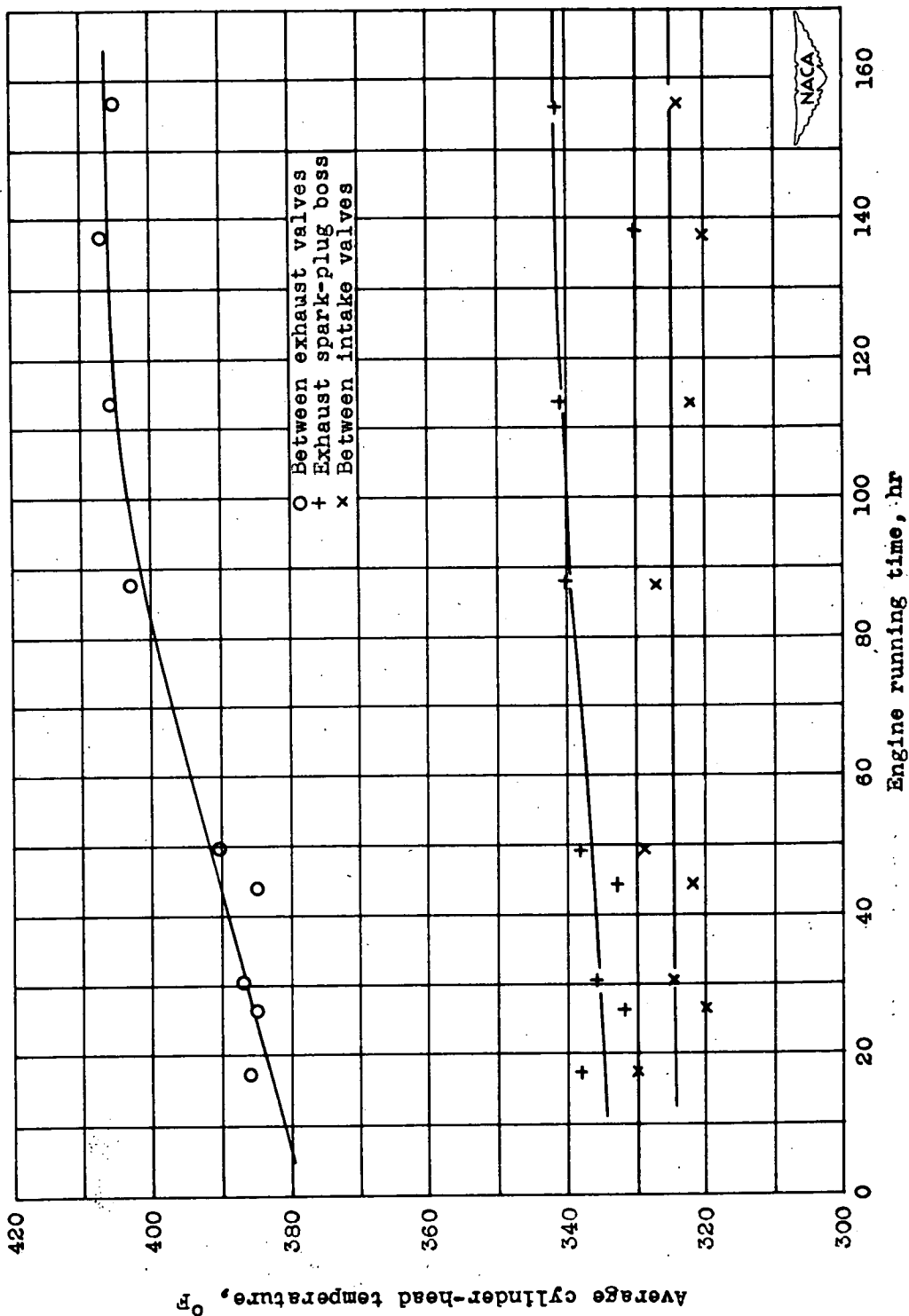


Figure 10. - Variation of average cylinder-head temperatures with engine running time. Engine speed, 2600 rpm; manifold pressure, 32 inches mercury absolute; charge flow, 1.57 pounds per second; fuel-air ratio, 0.095; carburetor-air temperature, 82° F; coolant, 30-70 ethylene glycol - water; coolant flow, 300 gallons per minute; average coolant temperature, 245° F; standard ignition timing; exhaust pressure, 29-30 inches mercury absolute; engine D.

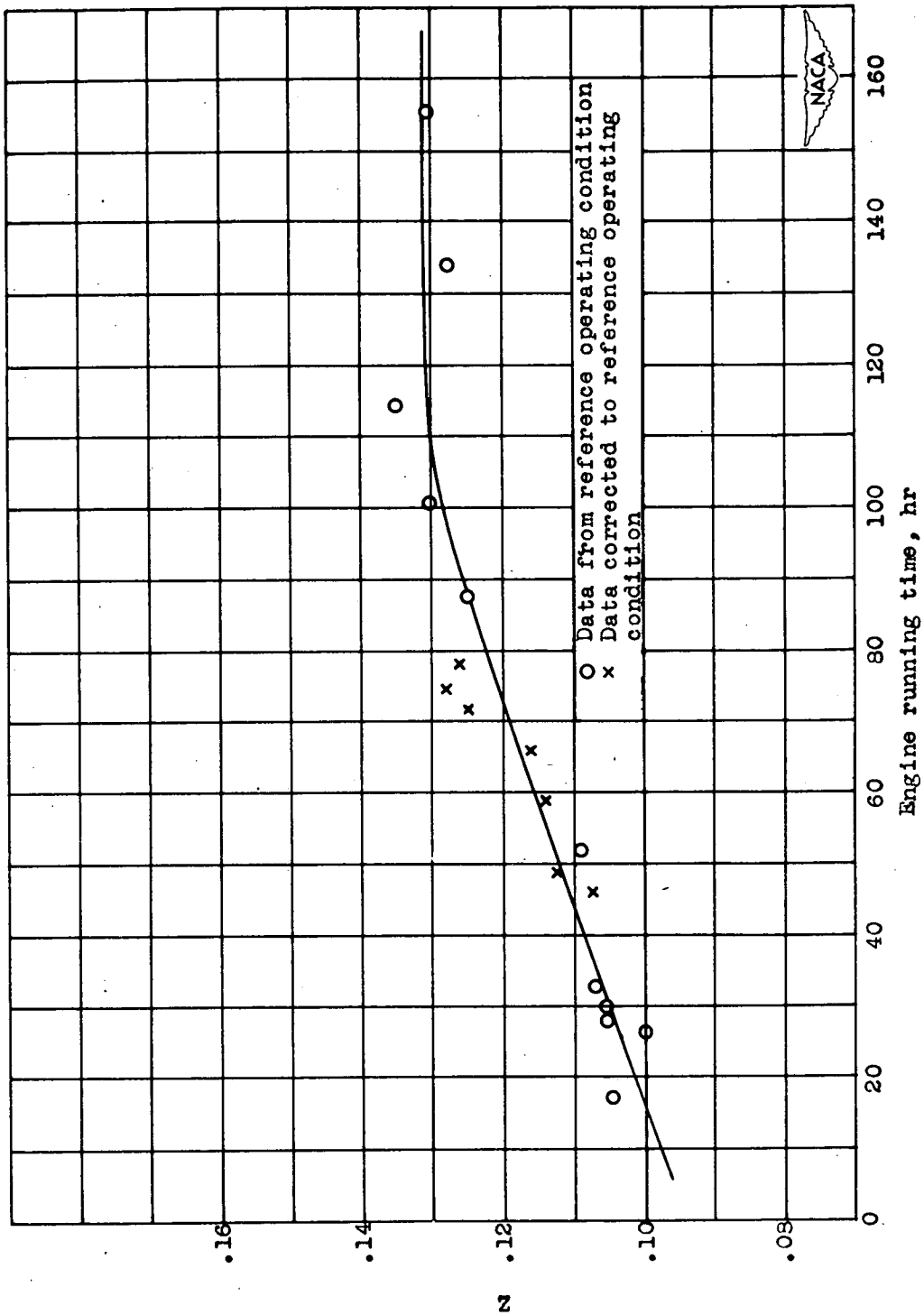


Figure 11. - Variation of factor Z with initial engine running time attributed to scale build-up in coolant passages of engine D.

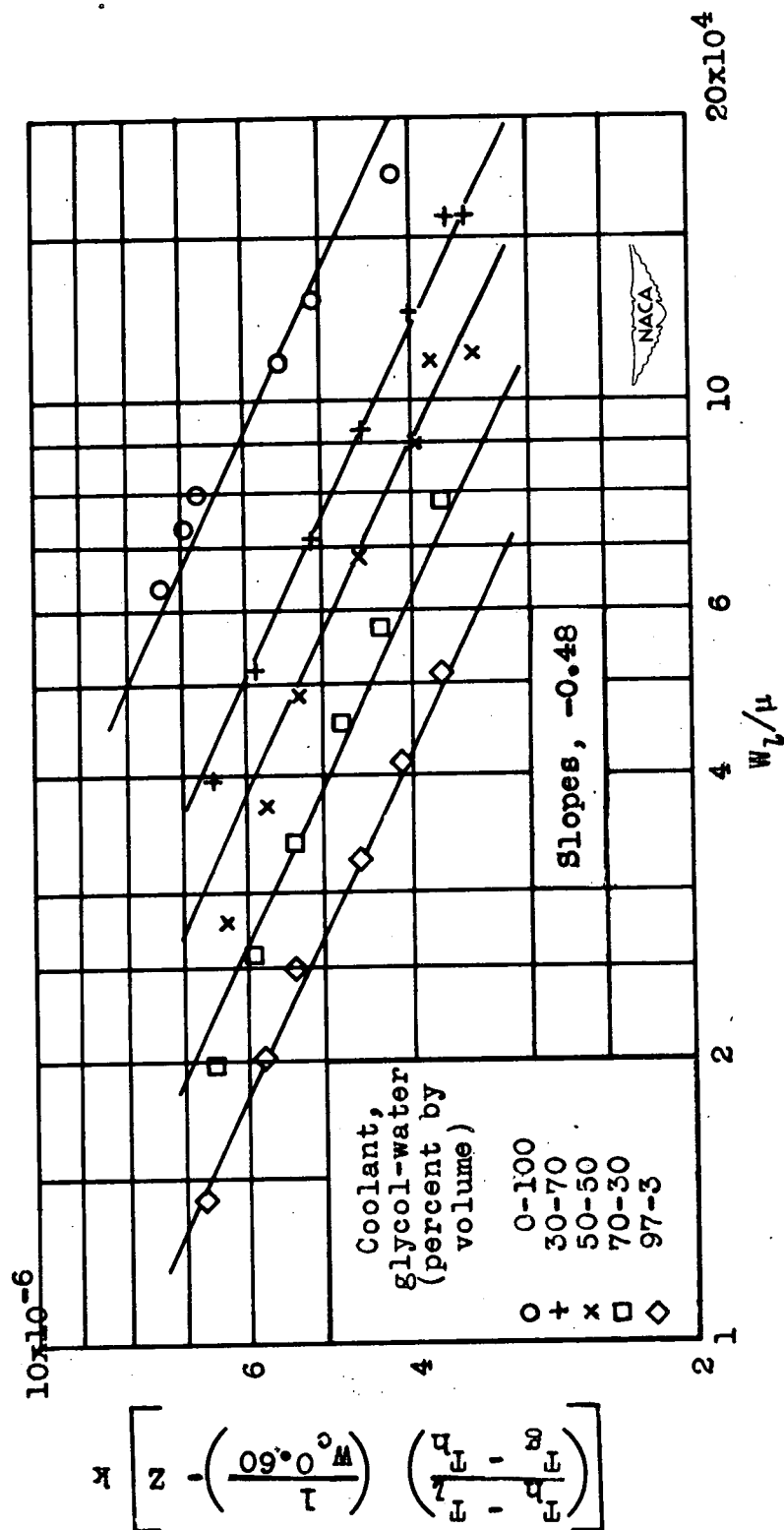


Figure 12. - Determination of exponent m on coolant-flow parameter W_l/μ from variation of $\left[\left(\frac{T_h - T_l}{T_g - T_h} \right) \left(\frac{1}{W_c^{0.60}} \right) - Z \right] k$ with W_l/μ for various coolants. Average coolant temperature, 245° F; engine D.

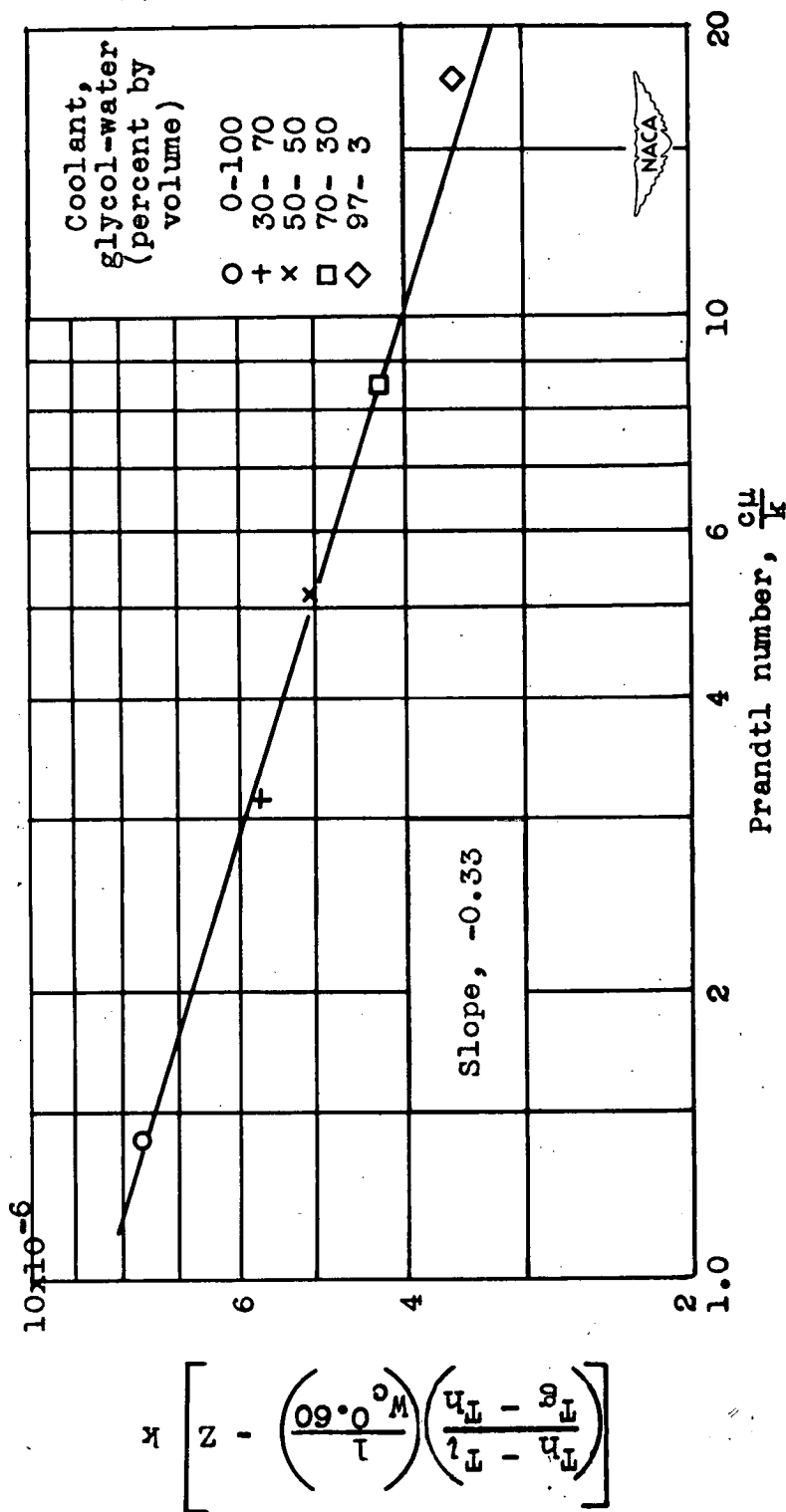


Figure 13. - Determination of exponent s on Prandtl number from variation of $\left[\frac{T_h - T_l}{T_g - T_h} \right] \left(\frac{1}{W_c^{0.60}} \right) - Z k$ with $\frac{c\mu}{k}$ as obtained from a cross plot of figure 12 at value for W_l/μ of 55,000. Average coolant temperature, 245° F; engine D.

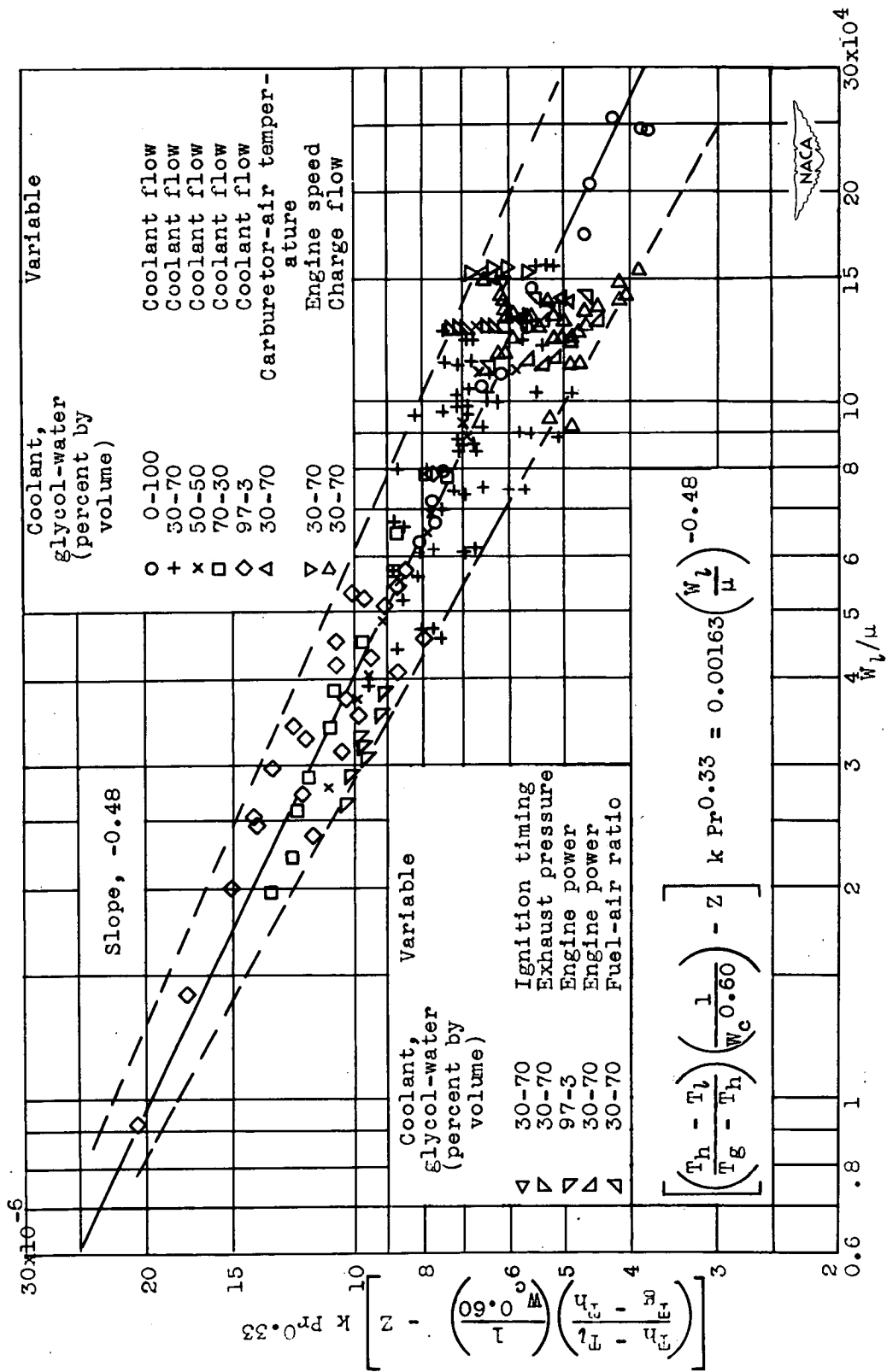
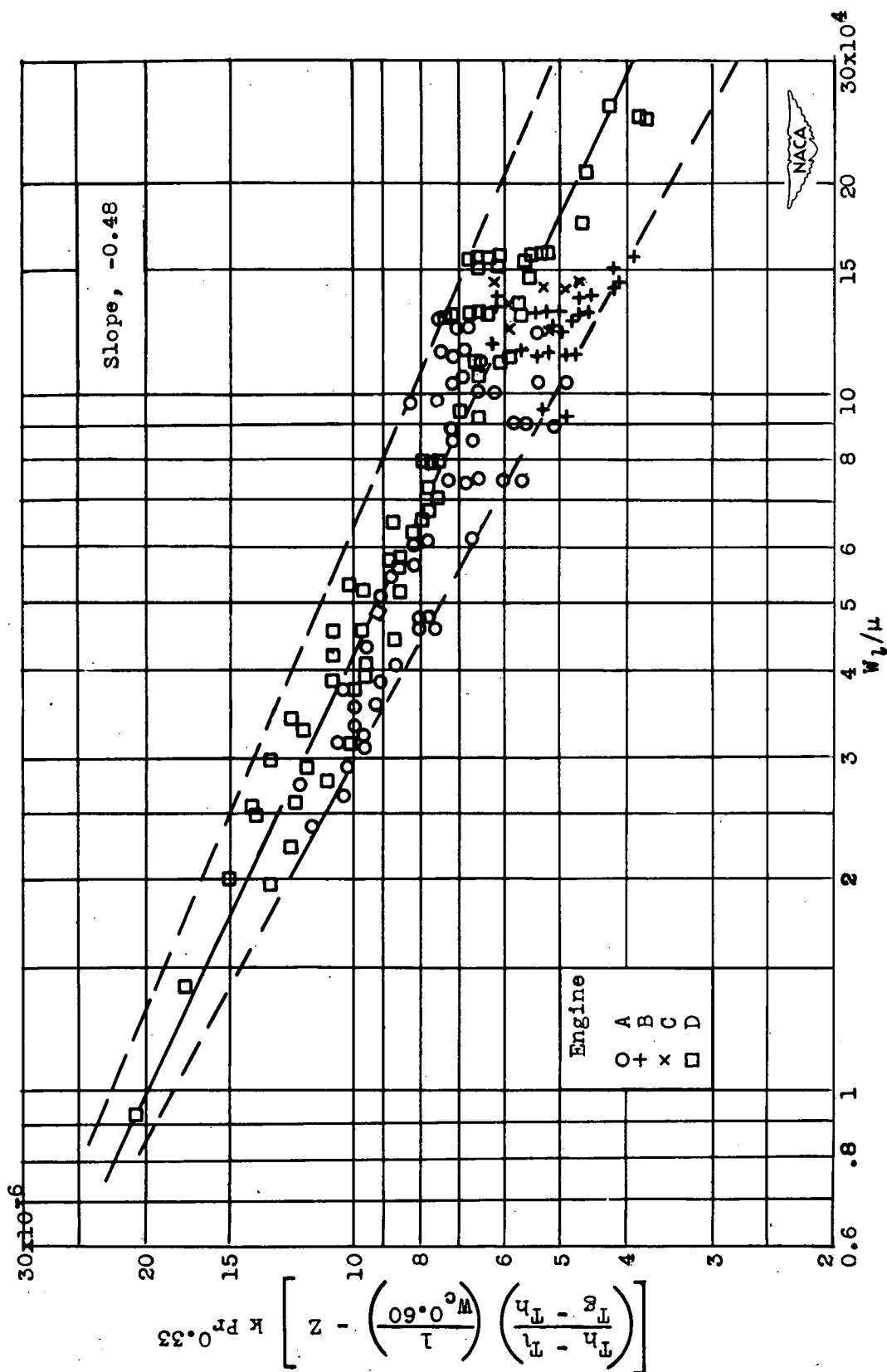


Figure 14. - Final correlation of cylinder-head temperatures based on equation (1).



(b) Data keyed for different engines.

Figure 14. - Concluded. Final correlation of cylinder-head temperatures based on equation (1).

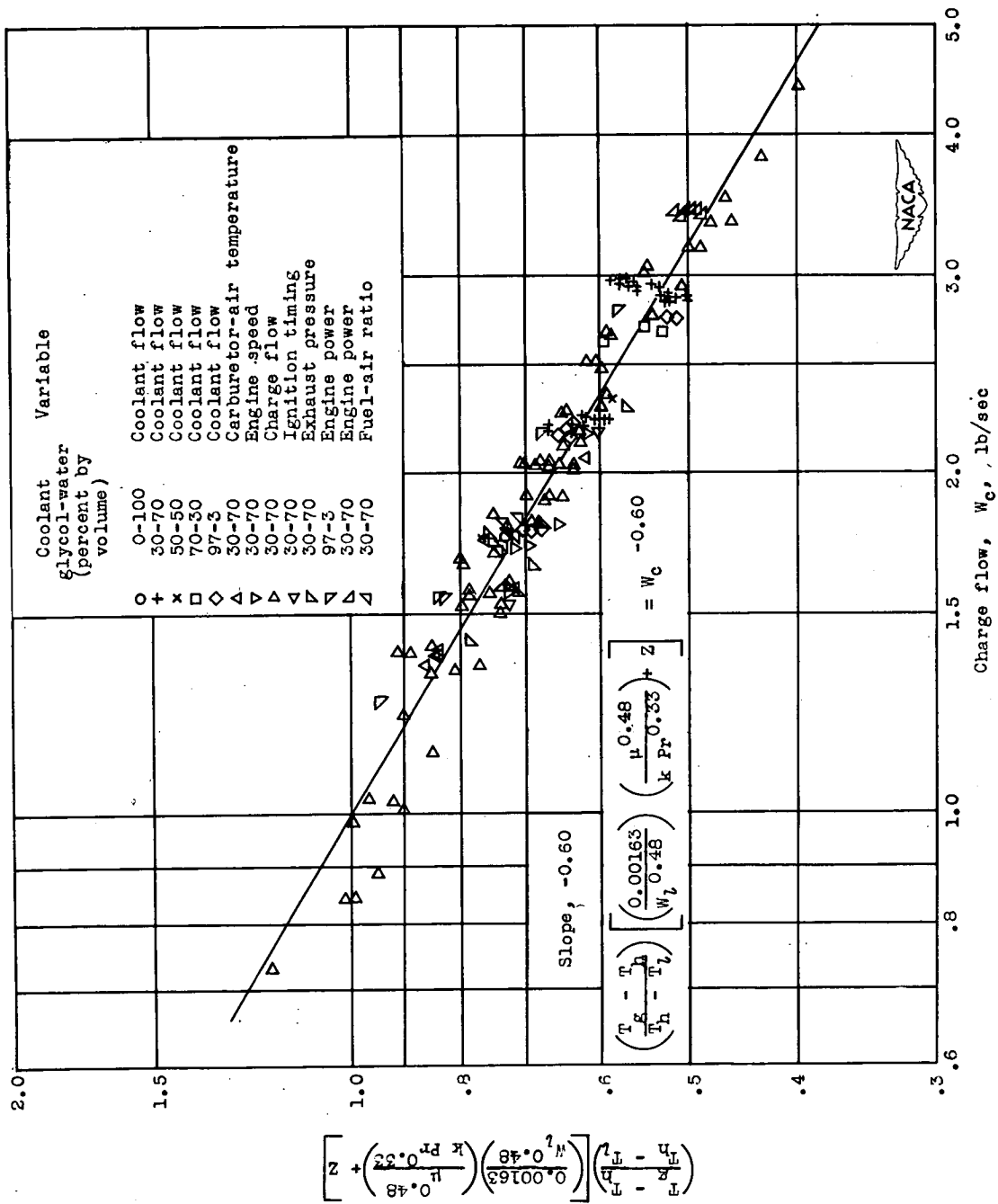


Figure 15. - Final correlation of cylinder-head temperatures based on equation (2).

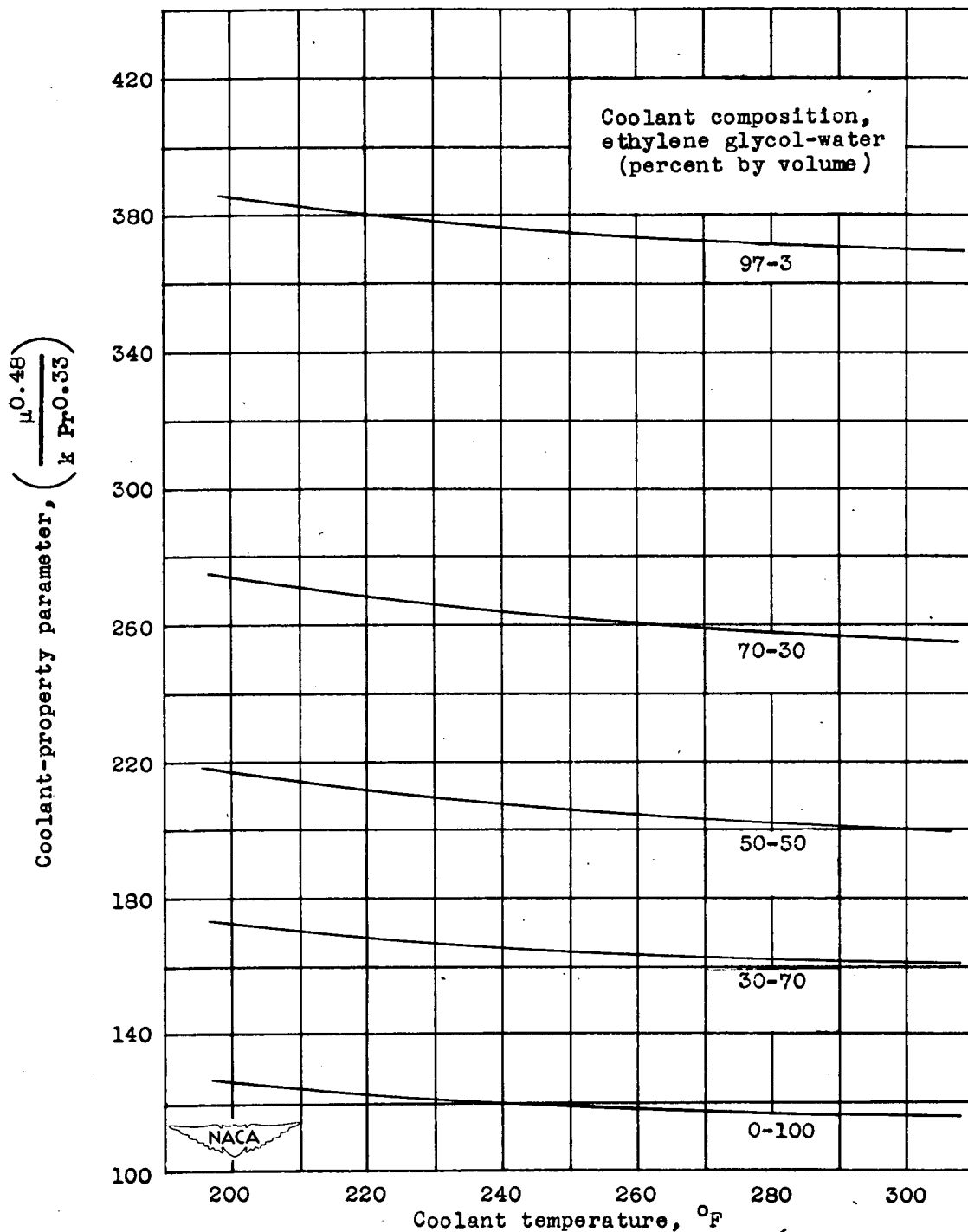


Figure 16. - Variation of coolant-property parameter $\left(\frac{\mu^{0.48}}{k \text{ Pr}^{0.33}} \right)$ with coolant temperature.

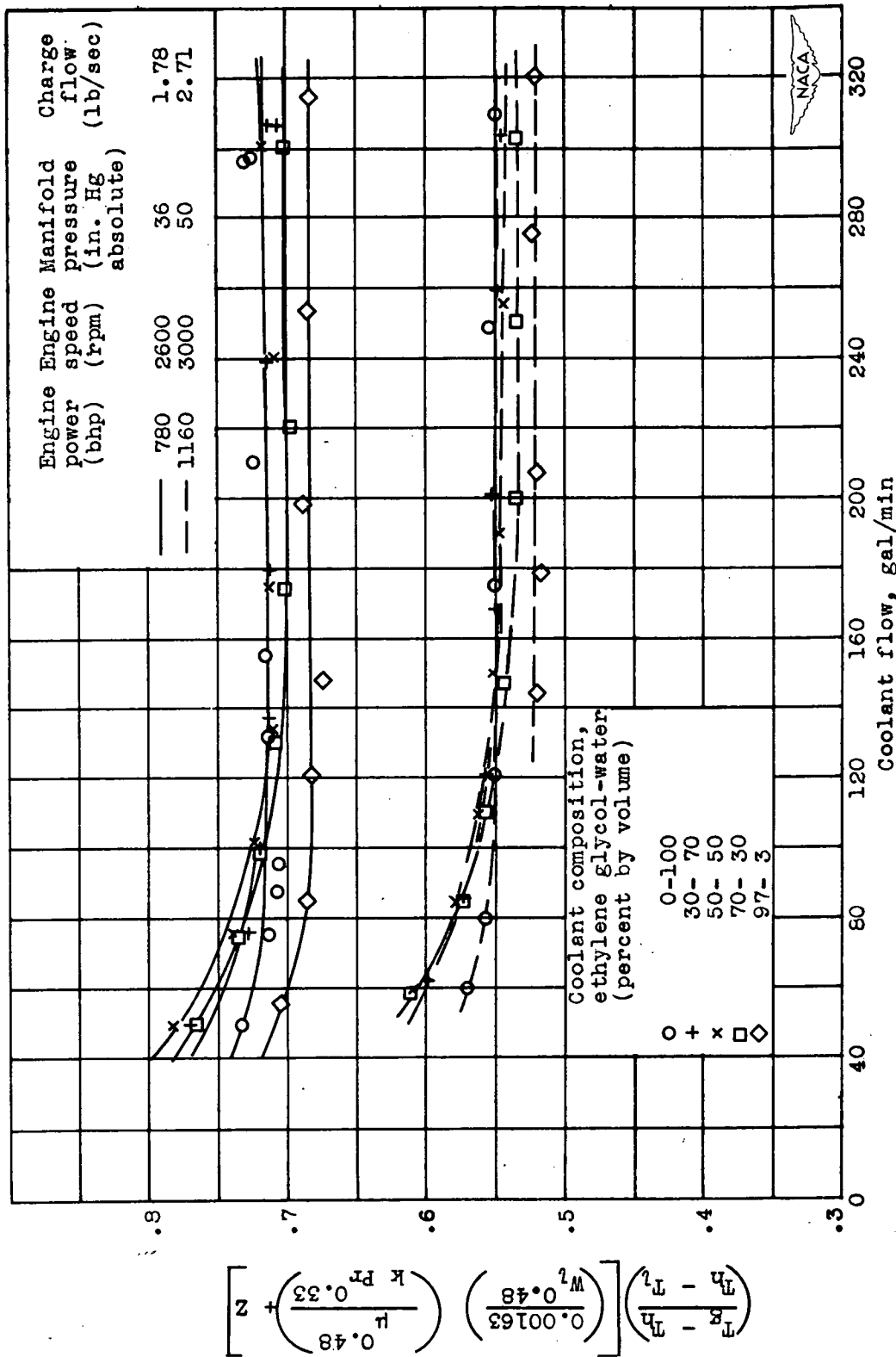


Figure 17. - Variation of $\left[\frac{T_g - T_h}{T_h - T_l} \right] \left[\frac{0.00163}{W_l 0.48} \right] \left[\frac{\mu}{k Pr 0.33} + Z \right]$ with coolant flow as indication of boiling. Average coolant temperature, 245° F; engine D.

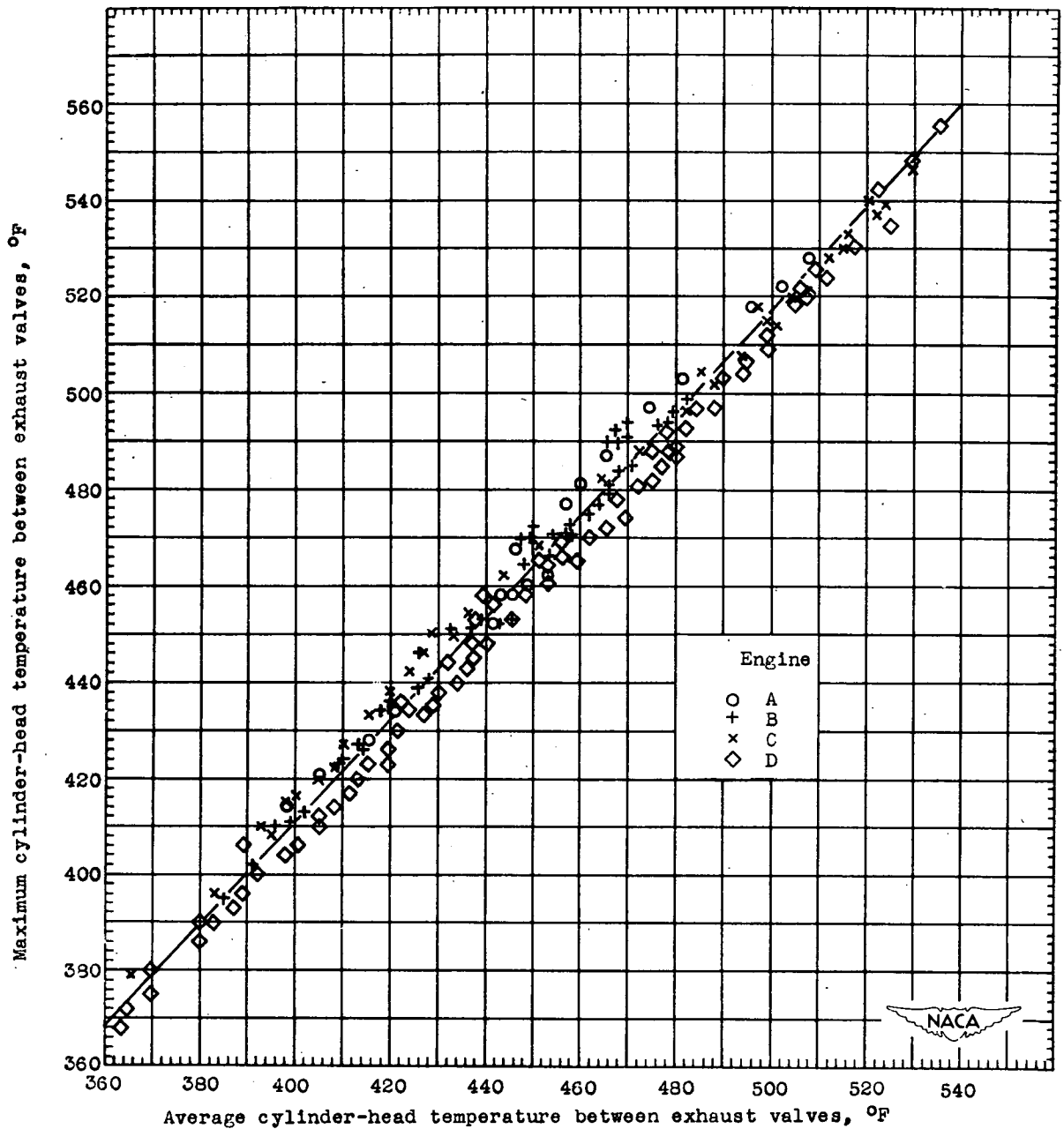


Figure 18. - Relation of average cylinder-head temperature between exhaust valves for 12 cylinders to hottest cylinder temperature measured at same location.